A STUDY OF FRICTION AND DETONATION IN GEOMETRICALLY SIMILAR ENGINES

BY

WARREN D .GABOURY
OTTO F. MEYER
PLEAS E. GREENLEE
JAMES W. SALASSI
AND
RAYMOND WIGGINS

A STUDY OF [illegible] AND
DEFORMATION IN GEOMETRICALLY
SIMILAR [illegible]

by

Warren D. Gaboury	Comdr. USN
Otto M. Meyer	Comdr. USN
Pleas E. Greenlee	Lt. Cdr. USN
James A. Salassi	Lt. Cdr. USN
Raymond Wiggins	Lt. Cdr. USN

Submitted in Partial Fulfillment of the
Requirements for the Degree of Master of Science
in Aeronautical Engineering from the
Massachusetts Institute of Technology

1950

Papers
G/2

May 19, 1950.

Professor J. C. Newell
Secretary of the Faculty
Massachusetts Institute of Technology
Cambridge 39, Massachusetts

Dear Professor Newell:

In compliance with the requirements for the Degree of Master of Science in Aeronautical Engineering from the Massachusetts Institute of Technology, we hereby submit a thesis entitled, "A Study of Friction and Detonation in Geometrically Similar Engines".

ACKNOWLEDGEMENT

The authors wish to acknowledge their indebtedness for the helpful advice and criticisms given by:

Professor C. F. Taylor

Professor E. S. Taylor

Professor P. M. Ku

Professor W. A. Leary

Mr. J. C. Livengood

Mr. D. S. Doremus

TABLE OF CONTENTS

LIST OF FIGURES

Appendix Figures

LIST OF TABLES.

ABBREVIATIONS AND SYMBOLS

BPSA	Best power spark advance, °BTC
°BTC	Degrees before top center
C_i	Speed of sound, conditions at inlet
C	Bearing clearance
D	Diameter
e	Volumetric efficiency (dimensionless)
E_c	Lower heating value of fuel, 19200 BTU/lb.
F	Fuel-air ratio
BSFC	Brake specific fuel consumption
ISFC	Indicated specific fuel consumption
GSE	Geometrically similar engines
h	Atmospheric pressure, "Hg.
BHP	Brake horse power
FHP	Friction horse power
IHP	Indicated horse power
$\dot{J}$	Mechanical equivalent of heat, 778 $\frac{ft.lb.}{Btu}$
K	Orifice flow coefficient (dimensionless)
ℓ	Characteristic dimension
$\dot{M}_f$	Mass fuel rate of flow, lb/unit time
$\dot{M}_a$	Mass air rate of flow, lb/unit time
M	Mach No. = s/c_i
MEP	Mean effective pressure, lb/in^2
BMEP	Brake MEP

FMEP	Friction MEP
IMEP	Indicated MEP
MMEP	Mechanical MEP
PMEP	Pumping MEP
N	Engine speed, revolutions per minute
n	Suction strokes per unit time
η_b	Brake thermal efficiency
η_i	Indicated thermal efficiency
η_m	Mechanical efficiency
O.N.	Octane number
θ	Crank angle, degrees
ρ	Density, lbs/ft^3
p	Pressure, in.Hg or psia as applicable
$\frac{dp}{dt}$	Differential of pressure with respect to time
p_a	Atmospheric pressure
p_e	Exhaust pressure
p_i	Inlet air pressure
Δp	Pressure drop across air orifice, "H_2O
psia	Absolute pressure, lb/in^2
psig	Gage pressure, lb/in^2
Re	Reynolds No. $= \frac{\rho s l}{\mu}$
r	Compression ratio
σ	Stress
s	Piston speed = 2SN, ft/min
S.A.	Spark advance
S	Piston stroke, ft.

T_a Air temperature before orifice, °F

T_{cyl} Cylinder head temperature, °F

T_f Fuel temperature °F

T_i Inlet air temperature, °F

T_o Oil temperature, °F

T_{wj} Water jacket temperature, °F

μ Absolute viscosity

V_f Flame speed, ft/sec

V_d Displacement volume, in^3

v Specific volume

ω Angular velocity

Y_1 Orifice expansion factor (dimensionless)

SUMMARY

The salient purposes of this thesis were:

1. To investigate the friction characteristics of the three M. I. T. geometrically similar engines.

2. To compare the relation between predicted FMEP characteristics and the actual observed FMEP characteristics.

3. To investigate the effect of variation of cylinder bore on detonation limits for constant piston speed and constant RPM.

The results of this study of friction characteristics of G. S. E. indicate that the simple friction theory was not completely adequate within the limitations of the test. The actual results show that FMEP varies inversely with bore at constant piston speed. As regards detonation, it was determined that knock limited BMEP and P_1 vary inversely with bore at either constant piston speed or constant RPM.

Within the knowledge of the authors, this represents the first experimental investigation of detonation characteristics of completely geometrically similar engines.

INTRODUCTION

During the last twenty years, or so, there has been developed a considerable amount of theory concerning the behavior of geometrically similar engines. It was the purpose of the study, reported upon herein, to investigate the variation of FMEP with piston speed for each of three geometrically similar engines. Further, it was desired to investigate the effect of variation of cylinder bore on detonation limitations, by determining the knock limited inlet pressure and knock limited BMEP for each of three geometrically similar engines, operating at the same piston speed, and, again, at the same RPM.

The engines employed in the study were designed by and built under the supervision of the Mechanical Engineering Department of the Massachusetts Institute of Technology. They are installed in the Sloan Laboratory of the Institute. Insofar as is known, these three engines are the only ones which have ever been built, which are completely geometrically similar in all respects. Details concerning the engines are to be found in Appendix A.

The only work which has been done previously on these engines was accomplished by Breed and Cowdery[1], Lobdell and Clark[2], and Mikel and McSwiney[3]. Experiments with similar

---

Superscript numbers refer to reference numbers.

cylinders of different size were conducted by Wunibald Kamm at the Deutschen Akademie der Luftfahrtforschung in 1939[4]. Though the cylinders used in these tests were essentially geometrically similar, the test set-ups which had to be used were not such as to maintain geometric similitude. They did correspond, approximately, in size and outlay to the cylinder sizes chosen, however. Comparisons of Kamm's data with the data of this report are included in a later portion.

In view of the limited amount of data previously taken on the M. I. T. geometrically similar engines, basic operating data was taken for each of the three engines as a prerequisite for the friction and detonation studies.

The study of engine friction was restricted to a determination of the FMEP of the assembled engines, as distinguished from determination of FMEP for major engine components. FMEP was determined by motoring the engine, and by firing and taking the difference of IMEP and BMEP.

The detonation studies were based on a single fuel; commercial "White Gasoline" which had an octane rating of 71.9 by the Motor Method and 78.4 by the Research Method. Data was taken for each engine operating at (1) a piston speed of 1200 feet per minute and (2) at 1000 RPM. Incipient detonation was used as a basis for determining knock limited inlet pressure and BMEP and was established by ear by one observer for the six inch bore engine. A Li Indicator, supplemented by ear, was used, by a single observer, in establishing detonation limits on the four and two and one-half inch bore

engines. The Li Indicator is described in Appendix B.

Much of the theory, which has been developed for geometrically similar engines, is based on dimensional analysis. Detailed development of applicable mathematical relations is to be found in Appendix C.

Theory predicts that IMEP is a function of R_e and M for geometrically similar engines. With identical inlet conditions:

$$IMEP = f(s, R_e)$$

Further, it is shown in Appendix C that, theoretically:

$$PMEP = f(s)$$

FMEP is normally considered to consist of two parts: (1) PMEP and (2) MMEP or mechanical friction mean effective pressure.

MMEP is, by definition, that portion of the so-called FMEP which arises from engine friction, as distinguished from the contribution of the pumping loss, i.e. PMEP. Consequently, the MMEP may be considered to be attributable to:

1. Coulomb friction
2. Viscous friction
3. Partial film friction

Presently, no technique of analysis is capable of predicting, or estimating, with any degree of reliability, the distribution of the engine friction among these three possibilities. Assuming, however, that all of the friction may be classified as either coulomb or viscous friction, a

theoretical treatment is made possible. Refer to Appendix C. Coulomb friction is a function of the "bearing" materials, largely, and may be taken as being independent of speed, load, and in general, of design and operating conditions. Therefore, since the materials used in the three geometrically similar engines, is the same, piece by piece, that portion of the MMEP which is attributable to coulomb friction will, in theory, be identical. Analysis, utilizing Petroff's Equation, leads to the conclusion that, for geometrically similar engines, with the same conditions of operation and at the same piston speed, the portion of the MMEP due to viscous friction, will be identical for each of the engines provided that the ratio of oil viscosity (absolute) to the bore of the engine (μ/ℓ) is the same for each of the engines. It follows, then, that theory indicates that the FMEP for geometrically similar engines will be the same, at the same piston speed, provided the Reynolds number-effect on IMEP is small.

Detonation in spark ignition is considered to be essentially a simultaneous combustion of the last portion of the charge in the cylinder to burn. It occurs when the flame speed is not sufficiently great to prevent the last portion of the charge from passing through its delay period before the normal progression of combustion has burned that portion. Experimental evidence to date tends to indicate that flame speed is a, more or less, linear function of Reynolds' number, but there does not seem to be, necessarily,

a one to one correspondence, i.e. flame speed does not appear to increase as fast as R_e. Fundamental information, relative to the manner of variation of flame speed with inlet Mach number, is lacking. If the engines be operated at the same inlet Mach number, which can be done by operating at the same piston speed, then the variation of average flame speed with R_e can be determined. At the same piston speed, with identical inlet conditions:

$$R_e \propto \ell$$

Consider an engine which has a bore twice that of a second engine. Then, the distance which the flame has to travel is twice as great, but if the flame speed is not twice as great, the larger cylinder will be more susceptible to detonation than the smaller.

EQUIPMENT AND PROCEDURE

The principal apparatus used consists of three single-cylinder, four stroke, geometrically similar spark-ignition engines of 2 1/2, 4, and 6 inch bore. All dimensions of the engines and their related equipment are in the same ratio as the bores.

Each engine is equipped with a suitable air orifice and surge tank, a vaporizing tank for mixing fuel and air, an exhaust surge tank, and cooling water header tank for water jacket coolant expansion. A rheostat controlled dynamometer with hydraulic scale permits brake measurements. Water jacket and oil heat exchangers with steam and water lines serve to control temperatures. Inlet and exhaust valves are used to control pressures. The air inlet throttle valves on the four inch and six inch bore engines are remotely controlled and operated by an electric motor. On the 2 1/2" engine the air inlet throttle valve is directly controlled manually and is capable of finer control. Related pumps, valves, piping, tubing, and manometers are shown in the diagram of the layout in Appendix A. A comparison of the engines is also presented. A detailed description has been given by C. F. Taylor[5].

Special equipment used included an MIT balanced pressure type indicator for taking indicator diagrams and a transfer machine for converting indicator diagrams to pressure-volume diagrams. This special equipment is described in Appendix

B. Associated minor equipment such as a fuel flow-bench for rotometer calibration, air compressors and reducing valves, strobotachometer, and stroboscopes were used as applicable.

The first part of the investigation was devoted to the gathering of basic operating data for the engines, e.g. best power fuel air ratio, best power spark advance, etc. Inlet pressure, exhaust pressure, inlet temperature, oil temperature, and water jacket temperature were held constant at 28" Hg, 32" Hg, 150°F, 150°F, and 180°F respectively to permit comparison of data. Data was also taken to show the variation of BMEP with inlet temperature and water jacket temperature for the 2 1/2" engine. One hundred octane fuel and special lubricating oil were used. They are described in detail in Appendix D. The engines were run at various piston speeds ranging from 400 to 1800 ft/min. After best power F was determined runs were made at the speeds and BPSA shown in the data sheets. Indicator cards were taken using the MIT indicator with the engines operating at the conditions specified above at best power F, BPSA and at various speeds. These cards were then transformed into pressure-volume diagrams and integrated by planimeter to give IMEP. In this analysis IMEP is defined as expansion stroke work minus compression stroke work divided by displacement volume. FMEP was determined by both the motoring and firing methods. In the latter, IMEP obtained from the p-v diagrams was reduced by the BMEP calculated from the dynamometer scale reading to give FMEP. No breakdown of motoring friction by parts was

attempted. A collection of the above described data provided a plot of IMEP, BMEP, and FMEP vs piston speed for each engine and an individual analysis could be made, as well as a comparison with the data of previous investigators.[1,2,3]

Detonation was the primary consideration in the second part of the investigation. The procedure followed was essentially the same as that of the friction investigation except that the Li indicator was used to determine detonation and no indicator cards were taken. A fuel of 71.9 O.N. (motor method) was found to be suitable to cause detonation in the three engines. In order to investigate detonation at the same piston speed in all engines it was necessary to supercharge the 2 1/2 and 4 inch engines. Inlet temperature was held constant at 180°F instead of 150°, while inlet pressure was used as a variable. This increased inlet temperature was necessary to prevent incomplete vaporization in the 2 1/2" engine vaporizing tank. Exhaust pressure, oil temperature, and water jacket temperature were held at their previous values. All engines were operated at a piston speed of 1200 ft/min and also at 1000 rpm. It should be noted that these speeds are identical for the 6" engine. It was found necessary to preset inlet pressure and to adjust spark advance until detonation occurred. This was necessary because of the difficulty experienced in controlling inlet pressure on the 4 and 6" engines with the remotely controlled air intake throttle valves, particularly with the

valves nearly closed. Further, engine equilibrium was disturbed more by varying inlet pressure than by varying spark advance to produce detonation. BPSA at best power F was determined for the 2 1/2 and 4" engines at three different inlet pressures. These pressures represented the low and high limits and an intermediate value of the supercharged inlet pressures used for the detonation study. One hundred O.N. fuel was used for these runs. From the results of these runs it was observed that BPSA was insensitive to inlet pressure within the range of interest and therefore previous BPSA data was used for the 6" engine.

In the 2 1/2" engine inlet pressure was varied from 32-42" Hg and spark advance was altered between 15 and 60 degrees BTC. Six fuel air ratios were investigated, all within the range of detonation. BMEP was determined for operation at a piston speed of 1200 ft/min and at 1000 rpm. Plots of BMEP and inlet pressure vs spark advance were made with F as a parameter. By entering these curves with BPSA for best power F, data was obtained for making a plot of BMEP and inlet pressure vs F.

The 4" engine inlet pressure ranged from 28 to 37" Hg and spark advance was varied from 10 to 60 degrees BTC. The same data was taken as for the 2 1/2" engine to give comparable BMEP and inlet pressure curves with F and spark advance as variables. Again six fuel air ratios were investigated. Runs for this engine were made at a piston speed of 1200 ft/min and also at 1000 rpm.

In the 6" engine F was changed as before and the spark advance spread used was 0-45 degrees BTC. No supercharging was necessary to cause detonation; therefore, inlet pressure ranged between 19 and 29" Hg. Since 1000 rpm and 1200 ft/min piston speed are identical for this engine, only one speed was run for comparison with the 2 1/2 and 4" engines. The same BMEP and inlet pressure plots were made as before using BPSA and best power F of .073. The same data was also taken in the case of the 6" engine for an inlet temperature of 150°F in order to show the effect of this variable on detonation.

After the above data had been compiled, a plot of knock limited BMEP and inlet pressure vs bore for a piston speed of 1200 ft/min and 1000 rpm at F of .0730 was made. A plot of knock limited IMEP vs bore for the same speeds was made by adding a value of FMEP (corrected for inlet pressure) to the values of knock limited BMEP.

One commonly used procedure to determine incipient detonation is to observe the first appearance of a protuberance on the oscilloscope dp-dt trace to the right of the maximum value. Because of the insensitivity of the protuberance to change in spark advance the above method was not sufficiently sensitive. Further, it was not possible to use a fixed height reached by the protuberance as a criterion for establishing detonation due to irregularity of the trace from one instant to the next. Consequently, the determination of the limiting condition

of detonation by the Li indicator was left to the decision of a single operator in the 2 1/2 and 4" engines. In view of the arbitrary definition of incipient detonation this procedure was considered more reliable.

Using a combination of oscilloscope indication, cylinder head temperature rise, and ear, it was possible to determine the high frequency pressure fluctuation associated with detonation and to reproduce incipient detonation with more accuracy than obtainable by ear alone. This is particularly true with the 2 1/2" engine.

It is interesting to note that only a three degree spark advance differential was found when incipient detonation was determined on the 4" engine by one observer using the Li indicator and by a different observer using ear alone. Greater spark advances were observed when the ear method alone was used.

It is believed that the point of incipient detonation in the 6" engine was accurately determined by ear alone due to the extreme sensitivity of knock to spark advance in this engine. Its full knock range, extreme knock to no knock, was observed in a span of approximately 3 degrees spark advance.

In both investigations certain precautionary measures and procedures were found to be applicable. A mean Reynold's number of 50,000 was originally assumed to facilitate air flow calculations. Air measurements were later

corrected for Reynold's number effect. This was found to be significant at slow speed and small air flow. For details of air flow measurements see Appendix E. Frequent calculations and checks of air and fuel flow were made to insure the accuracy of the F being used. Such steps were taken before, during, and immediately after each run. Ample time was allowed for warmup and for the establishment of steady operating conditions after altering any variable. Frequent checks were made to see that there were no air bubbles in the fuel inlet line or in the dynamometer hydraulic scale oil. The inlet vaporizing tank was inspected periodically for fuel condensation, particularly when running rich. Before and after each running period the spark advance mechanism was checked for slippage.

Pressures, temperatures, and speed were continually checked and controlled to promote accuracy. Pressures (except fuel and oil) and dynamometer scale reading were measured exclusively by manometers, the zero of which was noted before and after running. Temperatures were measured by thermocouples and calibrated potentiometers except for isolated fuel and air temperatures. The rotometers were calibrated for each of tho fuels used as described in Appendix F. No significant deviation was observed. Calibration curves will be found in Appendix F Figs. F-1, F-2, and F-3. Fuel temperature differential was not significant in this respect. Careful attention was paid to the operation of the MIT indicator, especially with regard to the establishment of the top center line.

Identical runs were frequently repeated on different days to check reproducibility of data. It was found that most runs checked within 1-2 per cent. Running plots were maintained as a check on reproducibility and to indicate desirable ranges of investigation and expansion of the schedule being followed.

Data readings were taken as nearly at the same time as possible throughout the study. This was made practicable by having at least two investigators present during each run.

RESULTS AND DISCUSSION

FRICTION

Fig. 1 illustrates that for the 2 1/2" engine, the BMEP, at best-power spark advance, increases with increasing fuel-air ratio until a fuel-air ratio is reached for which the BMEP must be a maximum. Further increase of fuel-air ratio results in a decrease of BMEP. The same trends are evidenced for the 6" engine in Figs. 3 and 4. Fig. 5 indicates that the best-power fuel-air ratio for the 2 1/2" and the 6" engines at piston speeds of 480 ft/min and 1200 ft/min, range from about .0715 to .0750. Accordingly, a value of F = .0730 was taken as being representative of the best-power fuel-air ratio. It should be noted that even though brake manometer readings were reproducible within one or two per cent, the curves are flat enough near the maximum that a one per cent variation could easily shift the apparent best-power fuel-air ratio within the range of the values indicated above. This conclusion is substantiated by References 1, 2 and 3, for these engines. In addition, Taylor and Taylor[6] state that, from theoretical considerations and from innumerable tests the best-power of a spark ignition engine, at best-power spark advance, will occur at a fuel-air ratio in excess of the chemically correct fuel-air ratio. For engines being fed a reasonably homogeneous fuel-air mixture which can be achieved by using a fuel vaporizing tank as was done in these tests, the best

power fuel-air ratio has been found to be lower than for engines fitted with carburetors; for the latter the best-power fuel-air ratio is about .0800[6].

In view of these considerations it was deemed unnecessary to duplicate this same data for the 4" engine. Accordingly, Fig. 2 shows the variation of BMEP at various piston speeds with spark advance at F = .0730 for the 4" engine. It is to be noted that best-power spark-advance increases with increasing piston speed, which is as expected.

Early in these investigations the question arose as to the magnitude of the effect of slight variations in the water jacket or inlet air temperatures. Fig. 6 shows the results of a special study of this question. A change of about 35 degrees in water jacket temperature is seen to give rise to a change of BMEP of approximately one-half pound per square inch. A variation in inlet air temperature of only three and one-half degrees produces the same change in BMEP. Consequently variations in water jacket temperature of as much as plus and minus five degrees will have a negligible effect on test results whereas the inlet air temperature must be held as close to the predetermined value as possible to insure reproducibility of results. These limitations are considered to be equally as applicable to the 4" and 6" G. S. engines.

In Appendix C, it has been shown in theory, for geometrically similar engines operating at identical inlet and exhaust conditions, that:

$$e = f(s)$$

This implies identical values of e for geometrically similar engines at the same piston speed. This conclusion is affirmed by Fig. 7 in which it is seen that e is substantially the same for all three engines tested, at the same piston speed.

Further, the theory described in Appendix C indicates that:

$$IMEP \propto e = f(s)$$

provided that the indicated thermal efficiency is a constant. If this be true, the IMEP's for these engines should be the same at the same piston speed. Fig. 8 indicates that the indicated thermal efficiencies for each of the three engines operating under the same conditions, are identical. Over a speed range from 480 ft/min to 1800 ft/min there is only a variation of four per cent in the value of indicated thermal efficiency. Therefore the assumption is justifiable.

Figs. 9 and 10, indicate the equality of the IMEP's for the three engines at the same piston speed.

Since ISFC $\propto 1/\eta_i$ for a given fuel and fuel-air ratio, the three engines should have identical values of ISFC at the same piston speed. This is seen to be substantially the case in Fig. 7.

In Fig. 10 have been plotted values of IMEP, BMEP and FMEP (motoring) and (firing) for the three geometrically similar engines at various piston speeds. The curves for FMEP (firing) represent the difference between the smooth curves for IMEP and BMEP.

Fig. 11 is a replot of the curves of Fig. 10 with the results of References 1, 2 and 3 superimposed thereon. In both cases, it is apparent that there exists a considerable difference between the values of FMEP's (firing). Consequently there is considerable difference among the engines in the values of BSFC as shown in Fig. 7 and the values of brake and mechanical efficiencies as shown on Fig. 8. BSFC is seen to increase with decreasing bore, and the brake and mechanical efficiencies are observed to decrease with decreasing bore. Similar results were observed by Kamm[4] in his tests with approximately geometrically similar cylinders. As stated earlier, simplified theory predicts that the FMEP's should be the same. The results and the theory disagree under the conditions of the tests. It is to be noted from Fig. 10, that both the motoring and firing FMEP's of the 2 1/2" engine are disproportionately higher in relation to the 4" engine than are the same values for the 4" engine in relation to the 6" engine. Fig. 11 shows that the increments of FMEP (firing) between the three engines, obtained by previous investigators[1,2,3], were more nearly uniform. Fig. 12 indicates that the motoring friction of the test set-ups used by Kamm[4] increased with diminishing bore in a very nearly unfform manner, wheras such uniformity is not evidenced by the M. I. T. geometrically similar engines. Even though the test set-ups used by Kamm were not strictly geometrically similar, they did bear a close relation to the size of the cylinders tested and should therefore be somewhat

indicative of the trend in FMEP (motoring) variation to be expected from the M. I. T. engines.

One possible explanation for this result is that there may exist some mechanical discrppancy within the 2 1/2" engine. In the early phase of these tests, the intake valve of that engine seized in the open position while the engine was running. Disassembly and inspection of the engine revealed a slight sticking of the exhaust valve and a defective main bearing. Data taken prior to this occurrence was discarded. Motoring tests should be conducted on engine components in an attempt to discover whether any such discrepancy does, in fact, exist.

Theoretical treatment of evaluation of power absorbed by bearings operating under conditions of full film lubrication has long been substantiated by numerous tests, and there is no reason to believe that the theory will not also apply in the case of these three geometrically similar engines. In attempting to analyze the reasons for the existence of the very large differences in FMEP (firing) between the three engines, the consideration of bearing and auxiliary friction may be disregarded, in view of the magnitude of their effect in relation to the magnitude of the effect to be expected of the piston and rings.

The simplified theory described in Appendix C implies that FMEP should be the same for all engines provided μ/ρ is the same. Inasmuch as oil viscosity is a function of temperature, a temperature must be specified

before the ratio μ/ϱ can be calculated. For the purposes of these tests a temperature of 250°F was picked as being somewhat representative, and oils were selected for each engine which would have the same μ/ϱ at that temperature. The oils used for the tests, were blends, as described in Appendix D. Viscosity tests, after blending, confirmed the constancy of μ/ϱ at 250°F among the three engines. Also, from these tests, the oils were found to have approximately the same viscosity index,and over a range of about 180°F to 350°F, the value of μ/ϱ for any one of the three engines did not differ radically from that of the other engines at any given temperature.

During the motoring tests, the engines were operating, more or less, within the temperature range specified above. The lubrication of the piston and rings, presumably, approximated viscous lubrication more closely than during the firing tests, and since the water jacket temperatures were held the same in all cases, after the engines cooled down, the cylinder wall temperatures became more nearly equal; therefore according to the theory, the FMEP's should have been the same. From Fig. 10 it is noted that the FMEP's (motoring) for the 4" and the 6" engines are essentially the same and the FMEP (motoring) for the 2 1/2" engine is much nearer the value of the 4" and 6" engines than it is during firing. By virtue of these facts, there seems to be some justification for the applicability of the theory under such conditions.

It is possible that during firing the mean engine cylinder wall temperature variation among the engines may be responsible for a part, at least, of the wide variance in FMEP (firing). For geometrically similar engines, operating with the same fuel-air ratio at the same piston speed, the heat transfer rate per unit area should be the same and approximately proportional to the ratio of the temperature difference across the cylinder wall (i.e. from inner to outer surface) to the wall thickness. With the coolant temperature being held the same for each of the three engines, the above indicates that the inner cylinder wall temperature should increase with increasing bore, and therefore the viscosity of the oil on the cylinder wall decreases with increasing bore. There would then be established a trend toward higher friction in the 2 1/2" engine and lower friction in the 6" engine.

Thus it is indicated that the choice of a single temperature to be used as a base for establishing the equality of the μ/ℓ 's for the three engines may not be justifiable. Livengood and Wallour[7] in a study of piston-ring friction conclude that increasing water jacket temperature decreases piston-ring friction. It may be presumed that this effect arises, largely, because of the decrease in viscosity with temperature increase. It is noted that the magnitude of this effect is appreciable. Changing water jacket temperature from 100°F to 150°F brought about a decrease in piston-ring mean effective pressure from about 7.7 psi to about 6.1 psi.

Accordingly, tests to determine the effect of cylinder water jacket temperature variation and oil viscosity variation on engine FMEP (firing) are considered to be desirable.

The theory is based on the assumption that all friction may be divided between coulomb and viscous friction. The lubrication conditions under which the piston and rings operate vary from coulomb to viscous to partial film, and are different for different parts of the piston travel. The parts are not separable or determinable at present and their proportions may vary from engine to engine. Therefore a part of the differences in FMEP's (firing) probably is due to those unknown variations.

There does appear to be the possibility, with further test experience, of establishing quasi-theoretical relationships by means of which the FMEP's of a group of geometrically similar engines may be predicted.

DETONATION

Results of the detonation investigation proper are shown in the curves of Figs. 14 through 25. Associated results, such as variation of cylinder head temperature and flame speed, may be found in Figs. 26 through 31.

BPSA at best-power fuel-air ratio for the supercharged detonation study at T_1 of 180°F was determined from Figs. 13 and 14 for the 2 1/2" and 4" engines respectively. A comparison with BPSA curves of Figs. 1 through 4, shows that T_1 and p_1 have no appreciable effect on BPSA within the ranges of p_1 and T_1 concerned. BPSA was found to be:

	2 1/2"	4"	6"
At s = 1200 ft/min	45	37.5	30
At N = 1000 RPM	30	32	30

At the same piston speed BPSA was found to vary inversely with bore, while at the same RPM, BPSA was approximately the same.

The knock limited MEP and p_1 of Figs. 15 through 20 are plots of recorded detonation data for the individual engines at various spark advances and speeds, with fuel-air ratio as a parameter. At constant speed, knock limited BMEP and p_1 are observed to decrease with increased spark advance for a given fuel-air ratio and to increase with fuel-air ratio at constant spark advance, except at lean fuel-air ratios where a reversal is apparent.

The variation in knock limited BMEP and p_i versus fuel-air ratio, with T_i as a parameter, for the 6" engine at BPSA and a piston speed of 1200 ft/min, is shown in Fig. 21. These curves are derived from Figs. 19 and 20. Both the knock limited BMEP and p_i are decreased by a 30° F increase in T_i, although at higher fuel-air ratios this effect on p_i is diminished. At a fuel-air ratio of .110, the p_i's for T_i of 150°F and 180°F are practically identical.

Plots of knock limited BMEP and p_i versus fuel-air ratio at BPSA and T_i of 180°F("S" curves) for all engines at a piston speed of 1200 ft/min and 1000 RPM are shown in Figs. 22 and 23 respectively. These curves were obtained from crossplots of Figs. 15 through 19, entering with BPSA corresponding to a fuel-air ratio of .073 for each engine. It is readily seen that for these conditions both knock limited BMEP and p_i vary directly with fuel-air ratio, except in the lean region. This tendency toward reversal at lean fuel-air ratios is most apparent for the 2 1/2" and 4" engines in Fig. 23. It shoul be noted that a reversal tendency also appears in the rich region for the 6" engine in Fig. 23.

From Figs. 15 through 19, knock limited BMEP and p_i, as a function of cylinder bore, are attainable for any set pattern of spark advances for the engines. Such a relation is presented in Fig. 24 for a given RPM and piston speed with a fuel-air ratio of .0730 and BPSA for each engine.

It is seen that knock limited BMEP and p_i vary linearly, but inversely, with bore at constant piston speed. At constant RPM, BMEP and p_i are observed to deviate slightly from this inverse-linear relationship at small bores. The above data is substantiated in Fig. 25, where a graphic comparison is made with results from Reference 4, which are at a different, but constant, piston speed. In order to compare these results on the same basis, it was necessary to estimate the IMEP's of the M. I. T. geometrically similar engines, since no indicator cards were taken during detonation runs. This was accomplished by adding the firing friction MEP's to the BMEP's observed for a piston speed of 1200 ft/min. Because the firing friction MEP's were determined at p_i = 28" Hg., a correction was applied to the FMEP's to account for the change in pumping loss due to variation in p_i in the detonation investigation. From experimental data, the magnitude of this correction was estimated to be of the order of $(28-p_i)/2$ inches of mercury or $(28-p_i)/4.06$ pounds per square inch. This correction represents only a maximum of about two per cent of the calculated IMEP.

It was interesting to note that, for any given engine, at incipient detonation, the cylinder head temperature attained a value which was constant for the same fuel-air ratio, independent of the spark advance and p_i. These values of cylinder head temperature at BPSA, are shown graphically for the three engines as a function of fuel-air ratio in Fig. 26. The values of cylinder head temperature at BPSA

and a fuel-air ratio of .073 at incipient detonation and at normal operation, are shown in Fig. 27. They are included as a comparative index to the heat transfer characteristics of the three engines under the conditions of investigation.

In order to understand the detonation characteristics of geometrically similar engines more fully, an investigation of the flame speed characteristics was undertaken. Flame speed is defined as the average velocity of the flame front, as measured by cylinder bore divided by the time interval from passage of spark to peak pressure, in units of feet per second. The measurement of the time interval was accomplished in two stages -- from passage of spark to top center, and from top center to peak pressure.

Since indicator diagrams were taken only at BPSA at a fuel-air ratio of .073, and as they were the only means available to estimate the time interval from top center to peak pressure, this operating condition was chosen to represent the flame speed characteristic. The data used was therefore readily obtainable from the data collected in the friction investigation. The degrees of crank angle from top center to peak pressure at BPSA and a fuel-air ratio of .073 was fairly constant at 14° (±2°) for all engines over the full range of speeds. Faired curves for BPSA versus piston speed for the three engines are shown in Fig. 28. With this information flame speed was calculated by the formula:

$$V_f = (B) \times (BPSA + 14°)\ N/2,\ ft/sec$$

where: B = cylinder bore, inches

N = RPM

Fig. 31 shows this relation of flame speed with RPM for the three engines. It is interesting to note that at the same RPM flame speed varies almost in direct proportion with bore. This alone would indicate that all three engines should have the same detonation characteristics at the same RPM.

With flame speed data available on the three engines, further computations were made to exploit the possibility of isolating Reynolds Number effect from Mach Number effect on flame speed. Reynolds Number, as used in the calculations, is defined as:

$$\frac{\rho_1 s' B}{\mu_1}$$

where ρ_1 = density of air at inlet conditions lbs/ft^3

s' = piston speed, ft/sec

B = cylinder bore, ft.

μ_1 = viscosity of air at inlet conditions lb/ft/sec

Mach Number is defined as:

$$s'/c_1$$

where s' = piston speed, ft/sec

c_1 = sonic velocity based on inlet conditions, ft/sec

Since inlet conditions were identical for all engines, piston speed is sufficient to represent Mach Number. In the M. I. T. geometrically similar engines, the gas velocities

through the inlet ports are the same for the same piston speed; therefore piston speed is also an appropriate index to the velocity of the entering mixture.

The above relations reveal that, in the case of the three engines, it is possible to obtain three values of piston speed for an assigned value of Reynolds Number. These three values of piston speed, each, correspond to a definite flame speed for each engine. By such a process, flame speed as a function of piston speed for various Reynolds Numbers was calculated and is shown in Fig. 30.

By a similar process, flame speed as a function of Reynolds Number, for various piston speeds, was also calculated and is shown in Fig. 29.

From these results it is seen that there exists the possibility of individual effects of Reynolds Number and Mach Number on flame speed. At low values of Reynolds Number (and Mach Number), flame speed is almost independent of Mach Number and dependent solely on Reynolds Number. At higher values of Reynolds Number (and Mach Number), flame speed tends to become more dependent on Mach Number and less on Reynolds Number.

To interpret the detonation characteristics of geometrically similar engines in the light of the results obtained in this investigation, it is advisable to resort to the auto-ignition theory of engine detonation described in Reference 6. Briefly, this theory can be explained as follows. As the flame front progresses across the cylinder,

the unburned charge is compressed by the hot, expanding gases and motion of the piston, until a critical combination of pressure and temperature is reached in the unburned end gas, at which time it auto-ignites. It has been established experimentally[8] that this critical pressure and temperature combination is dependent on the rate of compression, the composition (fuel, fuel-air ratio, fuel additives, etc.) of the unburned charge, and a time element referred to as "delay time".

In an attempt to analyze the detonation characteristics of the three geometrically similar engines with the data available, it is apparent that there is knowledge of only one of the above factors; namely, composition of the end gas. Flame speed, together with piston speed, can be used to evaluate the rate of compression of the end gas, only if the process is adiabatic or if the magnitude of heat flow to the end gas is known. Fig. 27 indicates that the heat transfer characteristics of the engines do not follow the laws of similitude at constant piston speed. At constant RPM, however, the pattern of cylinder head temperatures is one of increasing cylinder head temperature with increasing bore.

From considerations of flame speed and piston speed on the rate of compression of the end gas for a piston speed of 1200 ft/min in all engines, it is apparent that there is a basic advantage of small bores on knock limited output. Although flame speed increases with bore, it does not increase

in proportion to bore; hence, the smaller engine will have a lower combustion time and higher rate of compression. Also, the smaller engine is operating at a higher RPM, which results in a higher rate of compression. The basic disadvantage of the 6" engine is magnified by a higher operating temperature, whereas, in a comparison of the 2 1/2" and 4" engines, the advantage of operating temperature, using cylinder head temperature as an index, favors the 4" engine.

At an RPM of 1000, it is observed that flame speed varies directly as bore, resulting in equal rates of compression of the end gas in all three engines. As the RPM is the same in this case, equal rates of compression result due to this factor. The significant factor that is needed to account for the variation of knock limited BMEP with bore is therefore the rate of heat flow to the end gas during the compression process. If the cylinder head temperature is accepted as an index to this rate of heat flow, the variation of output with bore may be explained. In light of other important factors and their relative influence, such as radiation across the flame front and auto-ignition characteristics of a particular fuel, etc., it would be rather unwise to pursue such an argument in a general case.

CONCLUSIONS AND RECOMMENDATIONS

Under the conditions of the tests reported in these investigations, the results obtained give rise to the following conclusions:

Friction

1. The best-power fuel-air ratio for the three engines is approximately .0730 and largely independent of speed.
2. The theory, which predicts that the IMEP's, indicated thermal efficiencies, indicated specific fuel consumptions, indicated specific air consumptions, and volumetric efficiencies of geometrically similar engines will be the same at identical operating conditions at the same piston speed, is confirmed by the tests.
3. In relation to the FMEP's (firing) there appears to be disagreement between theory and the results within the conditions of the tests.
4. In regard to the FMEP's (motoring) the theory is more nearly applicable.
5. Selection of a single temperature to be used as a base for establishing the equality of the μ/ρ ratios for the three engines may not be justifiable.

6. The scope of these tests was not broad enough to establish or to refute the applicability of the theory in regard to the FMEP of geometrically similar engines.

Additional tests to determine the effect of variation of cylinder water jacket temperature and variation of oil viscosity are recommended.

7. Motoring tests of engine components should be made to determine whether or not there exists a mechanical discrepancy in the 2 1/2" engine.

8. There appears to be the possibility, with further test experience, of establishing quasi-theoretical relationships by means of which the FMEP's of a group of geometrically similar engines may be predicted.

Detonation

9. The results of this investigation indicate that the knock-limited BMEP and p_1 vary inversely with cylinder bore in geometrically similar engines at either the same piston speed or the same RPM.

10. There is indication of individual effects of Reynolds Number and Mach Number on the average speed of propagation of a flame front in a cylinder.

11. In order to correlate the detonation characteristics more fully with variables whose effects on detonation have been determined experimentally, the heat transfer characteristics of geometrically similar engines should be thoroughly investigated.

12. To supplement this study of detonation, it is recommended that a similar procedure be followed at other combinations of RPM and piston speed, and using various fuels.

REFERENCES

1. MIT Thesis 1949 "A Study of the Laws of Similitude Using the Six Inch G. S. E.", E. Breed and J. R. Cowdery.
2. MIT Thesis 1949 "A Study of the Principle of Similitude Using the Four Inch G. S. E.", L. W. Lobdell and R. E. Clark.
3. MIT Thesis 1949 "A Study of the Laws of Similitude Using a 2 1/2" G. S. E.", F. G. Mikel and D. D. McSwiney.
4. "Results of Experiments with G. S. Cylinders of Different Size, and Conclusions for the Development of Aircraft Engines" W. Kamm, translated from German by G. H. Dibelius July 1949.
5. ASME paper 49-A-116 presented Nov. 27-Dec. 2, 1949, meeting "The Effect of Size on the Design and Performance of Internal Combustion Engines", C. F. Taylor.
6. "Internal Combustion Engine" Revised Edition 1948, International Textbook Company, C. F. Taylor and E. S. Taylor.
7. NACA TN 1249 "A Study of Piston-Ring Friction, J. C. Livengood and C. Wallour.
8. SAE Preprint "The Ignition of Fuels by Rapid Compression" presented at SAE meeting Jan. 9-13, 1950, C. F. Taylor, E. S. Taylor, J. C. Livengood, W. A. Russell, W. A. Leary.
9. NACA Report 4J06 Nov. 1944 "A Study of Piston and Ring Friction", W. A. Leary and J. U. Jovellanos.
10. Mechanical Engineering, Mar. 1933 "A New High Speed Indicator", E. S. Taylor and C. S. Draper.

11. NACA TN 1896 June 1949, "Improvement of the Accuracy of Balanced Pressure Indicator", J. C. Livengood.

12. Journal of Aeronautical Sciences, Oct. 1949, "A New High Performance Engine Indicator of the Strain-Gage Type", C. S. Draper and T. T. Li.

13. "Measurement of Air Flow by Means of ASME Square Edged Orifice with Flange Taps", Sloan Laboratory, MIT, W. A. Leary.

14. "Theory of the Flowrator", Catalog Section 98-A, 1947 Fischer and Porter Co., Hatboro, Pa.

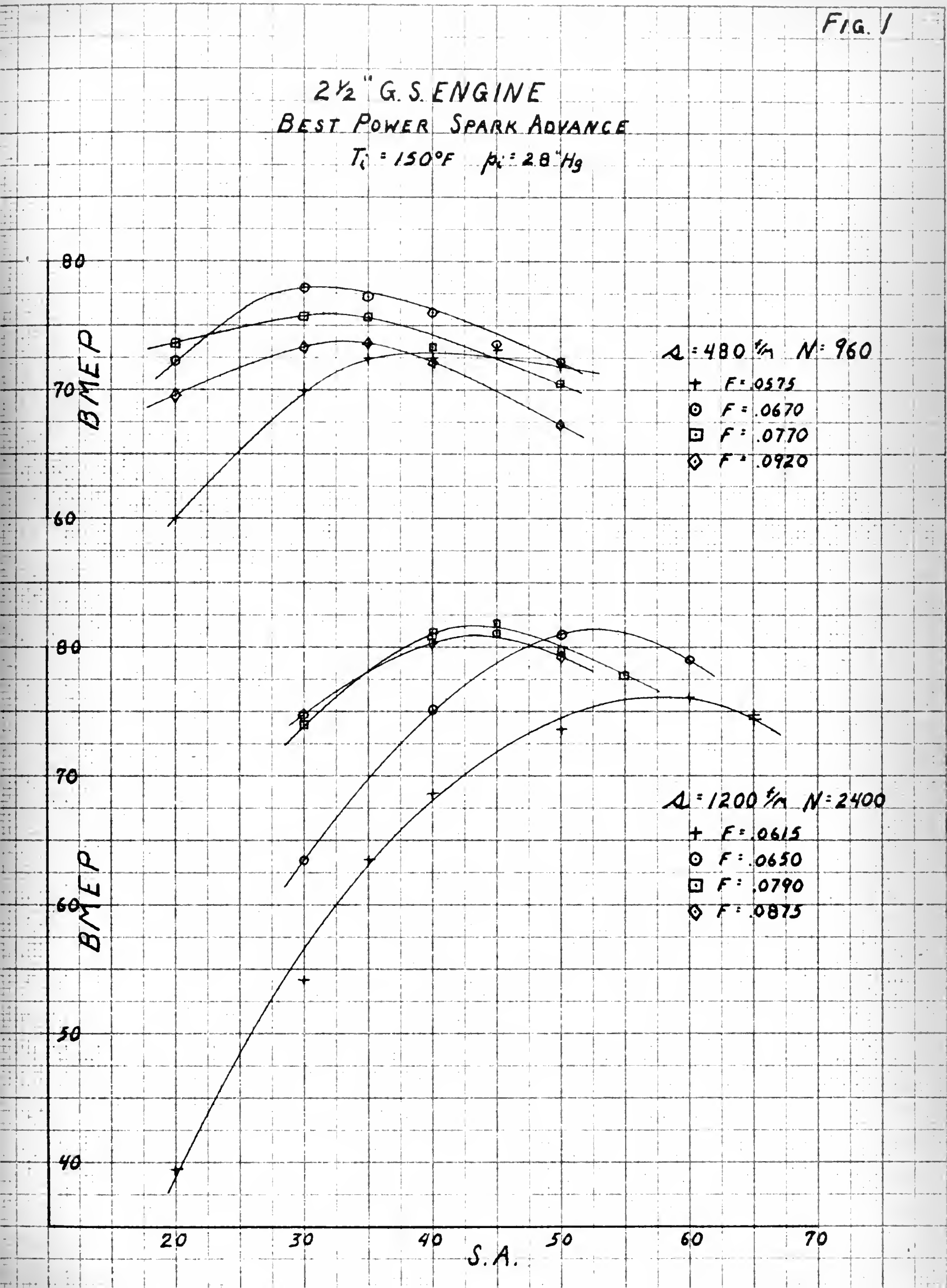
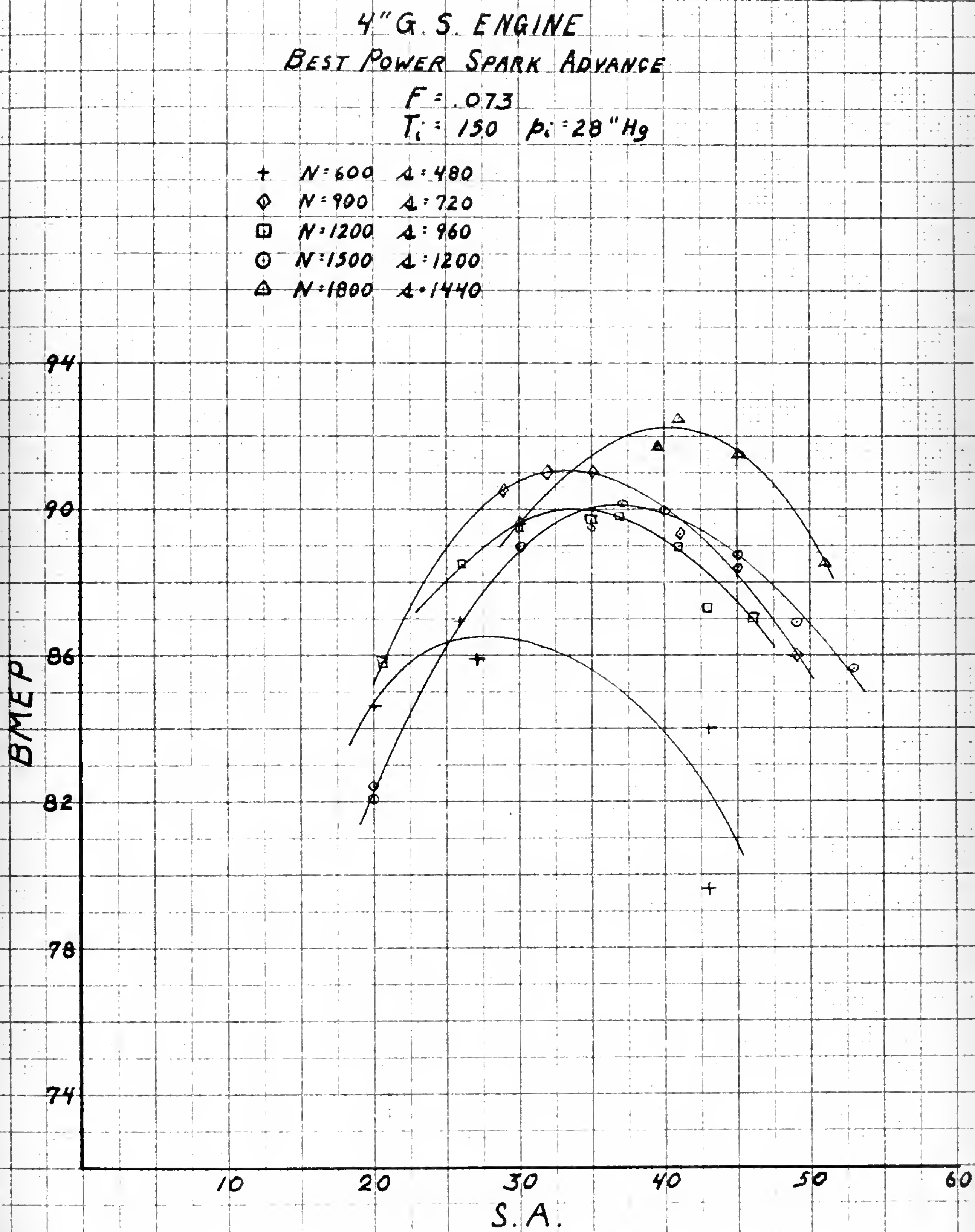


FIG. 1
2½" G.S. ENGINE
BEST POWER SPARK ADVANCE
T_i = 150°F p_i = 28" Hg
BMEP
S.A.
s = 480 ft/m N = 960
+ F = .0575
⊙ F = .0670
⊡ F = .0770
◇ F = .0920
s = 1200 ft/m N = 2400
+ F = .0615
⊙ F = .0650
⊡ F = .0790
◇ F = .0875
80
70
60
50
40
20
30
40
50
60
70

FIG. 2



4" G. S. ENGINE
BEST POWER SPARK ADVANCE
F = .073
Ti = 150 pi = 28" Hg
+ N = 600 ∆ = 480
◇ N = 900 ∆ = 720
□ N = 1200 ∆ = 960
○ N = 1500 ∆ = 1200
△ N = 1800 ∆ = 1440
BMEP
94
90
86
82
78
74
10
20
30
40
50
60
S. A.

FIG. 3

6" G.S. ENGINE
BMEP VERSUS SPARK ADVANCE
$T_i = 150\,°F$
$P_i = 28$" Hg

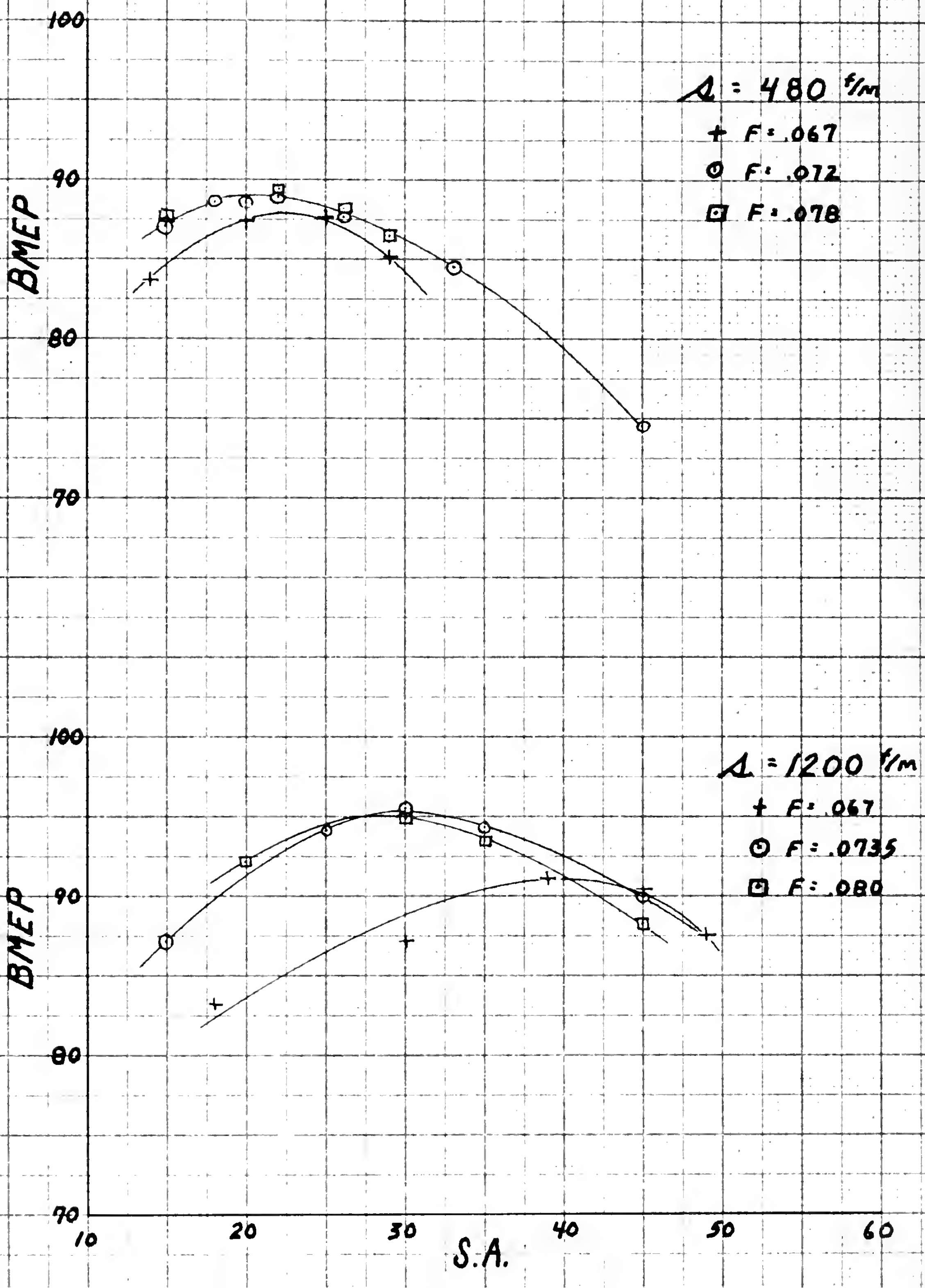
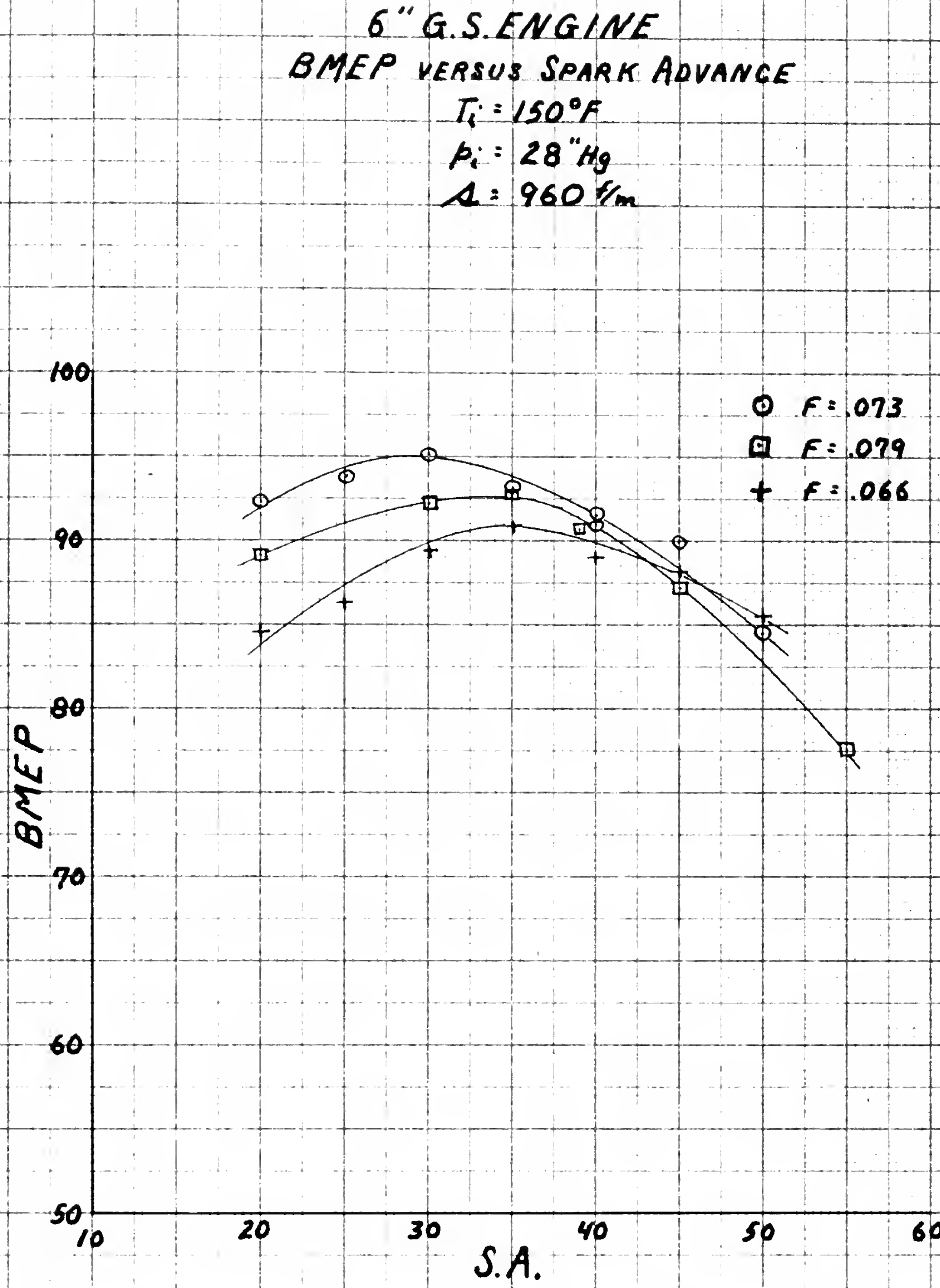
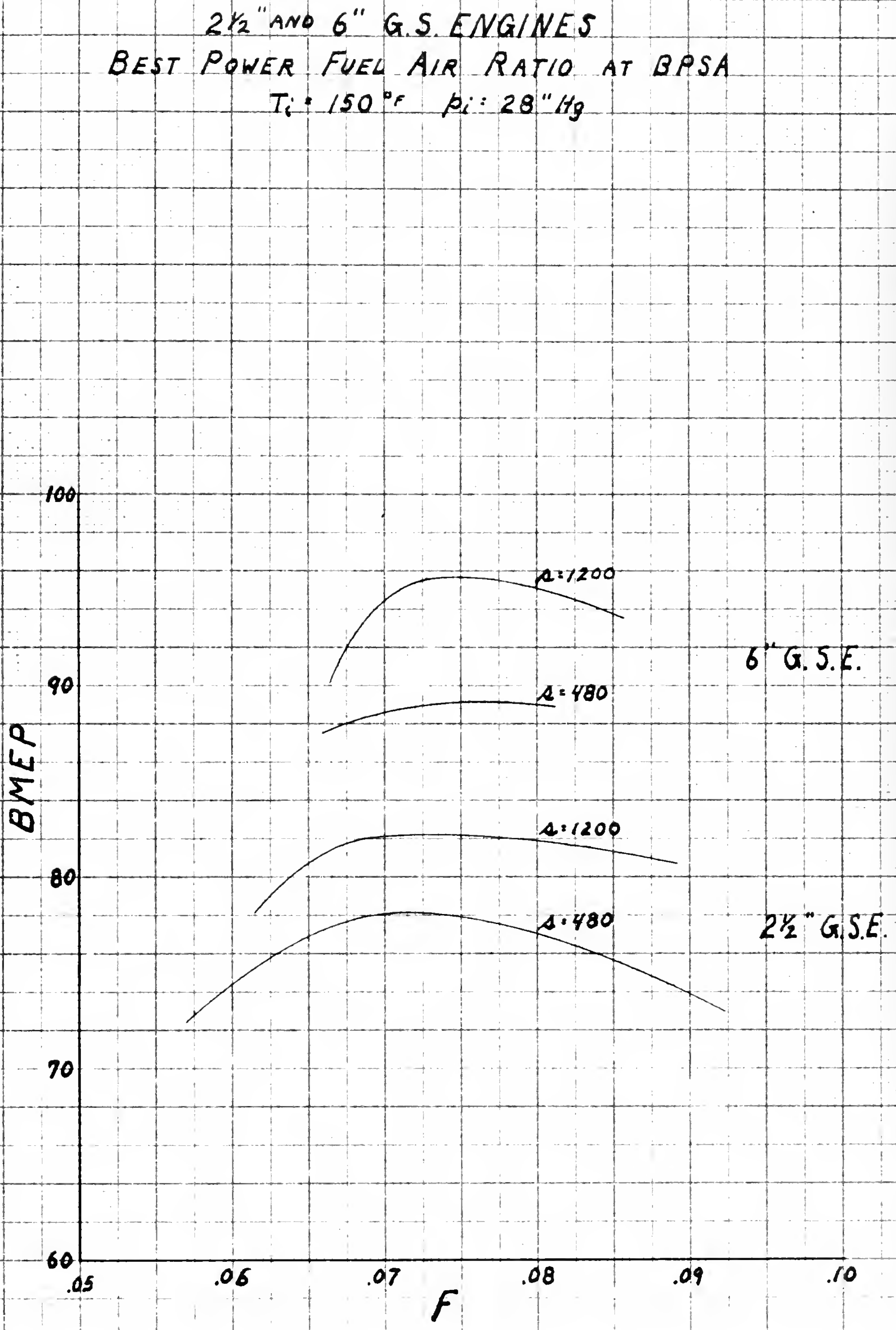


FIG. 4



6" G.S. ENGINE
BMEP VERSUS SPARK ADVANCE
T_i = 150°F
p_i = 28" Hg
s = 960 ft/m
F = .073
F = .079
F = .066
100
90
80
70
60
50
BMEP
10
20
30
40
50
60
S.A.

FIG. 5



2½" AND 6" G.S. ENGINES
BEST POWER FUEL AIR RATIO AT BPSA
T_i = 150°F p_i : 28" Hg
BMEP
100
90
80
70
60
.05
.06
.07
.08
.09
.10
F
s:1200
s:480
6" G.S.E.
s:1200
s:480
2½" G.S.E.

2½" G. S. ENGINE

VARIATION OF BMEP WITH T_{wj} AND T_i

$A = 1200$ $N = 2400$

$P_i = 28''$ Hg BPSA $F = .073$

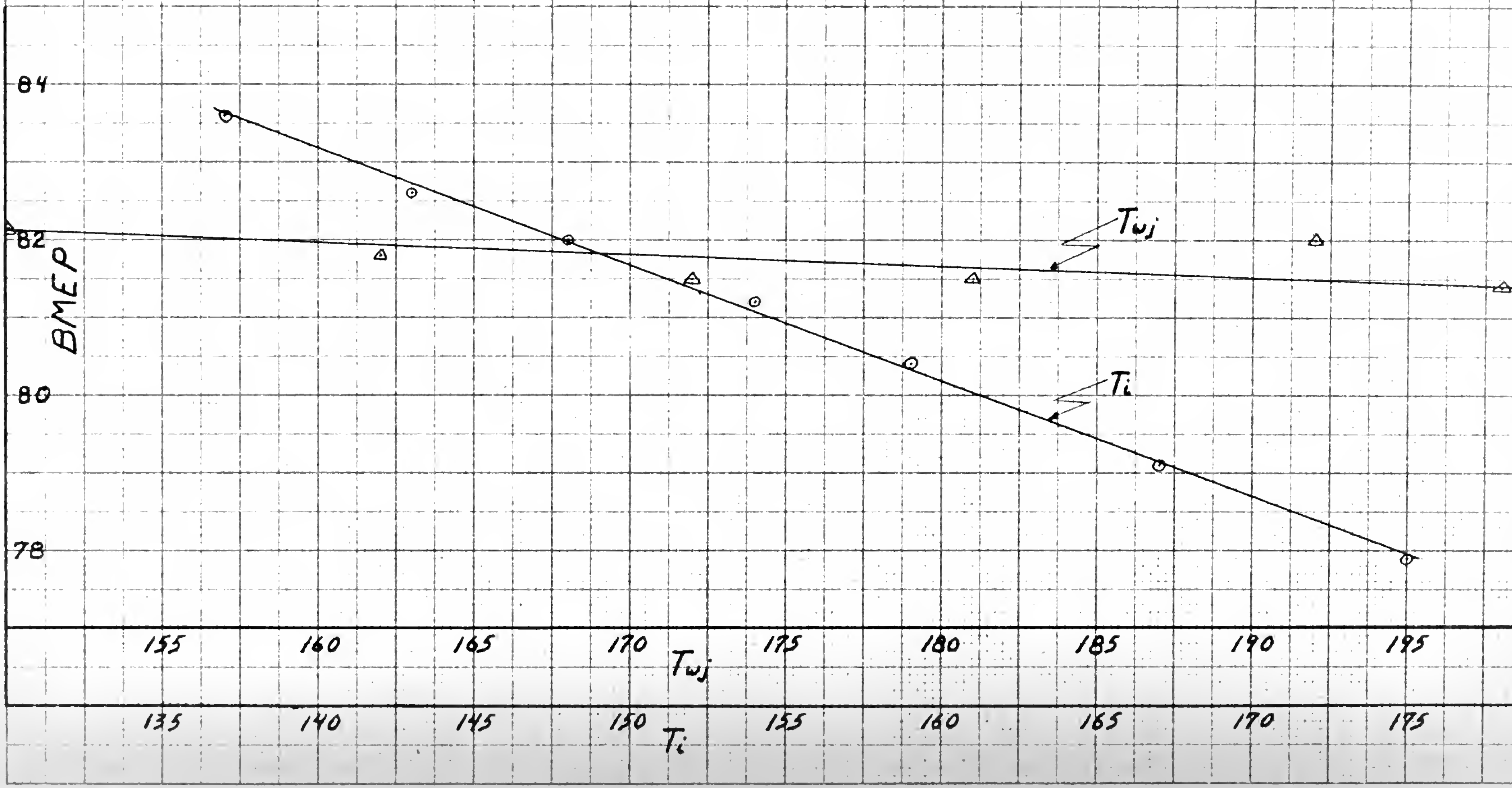
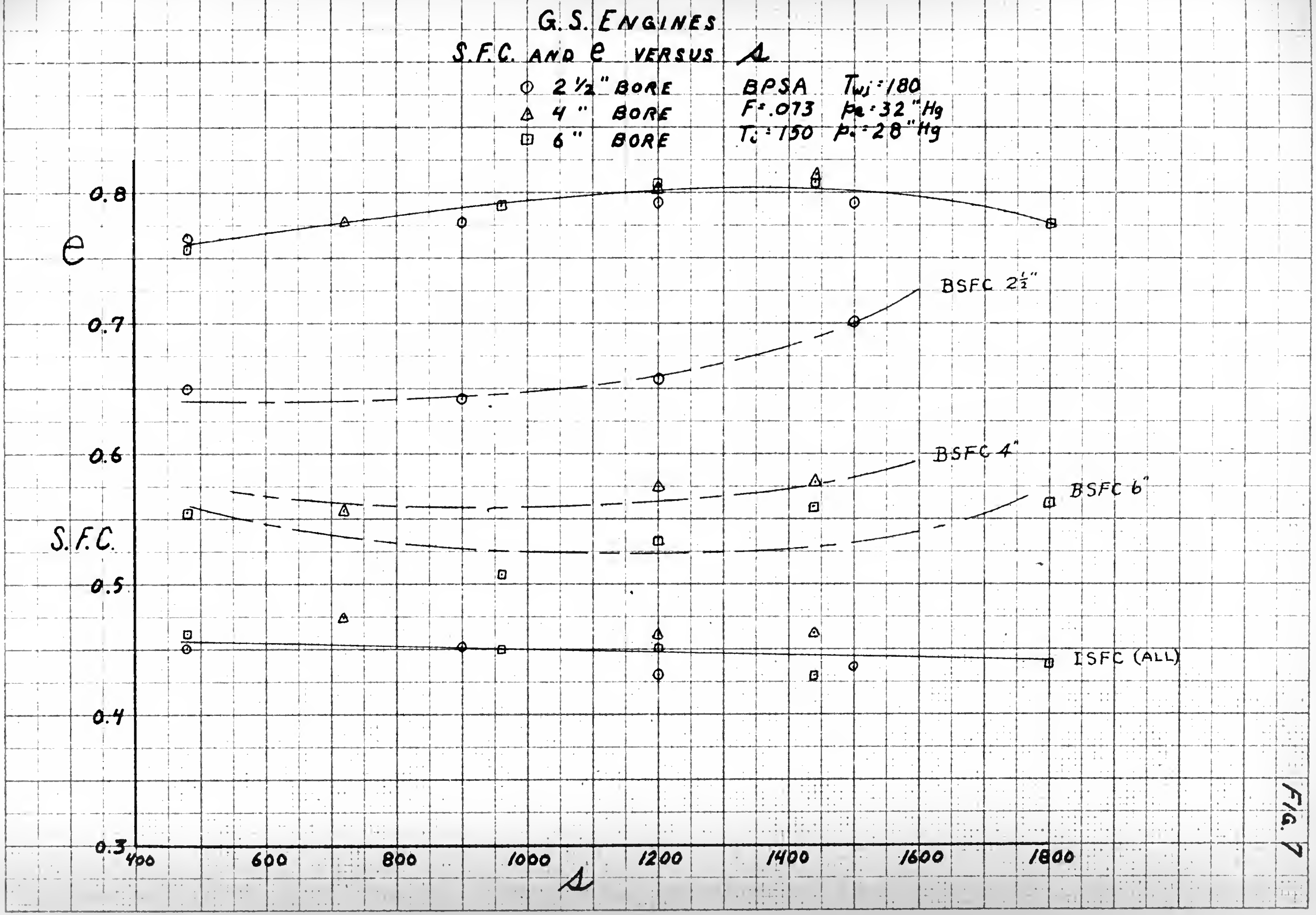


FIG. 6



G.S. ENGINES
S.F.C. AND e VERSUS s
○ 2 1/2" BORE
△ 4" BORE
□ 6" BORE
BPSA
F = .073
Ti = 150
Twj = 180
Pe = 32" Hg
Pi = 28" Hg
BSFC 2 1/2"
BSFC 4"
BSFC 6"
ISFC (ALL)
e
S.F.C.
0.8
0.7
0.6
0.5
0.4
0.3
400
600
800
1000
1200
1400
1600
1800
s

FIG. 7

G. S. ENGINES

VARIATION OF EFFICIENCIES WITH PISTON SPEED

BPSA   F = .073
T_i = 150 p_i = 28" Hg
T_{wj} = 180 p_e = 32" Hg

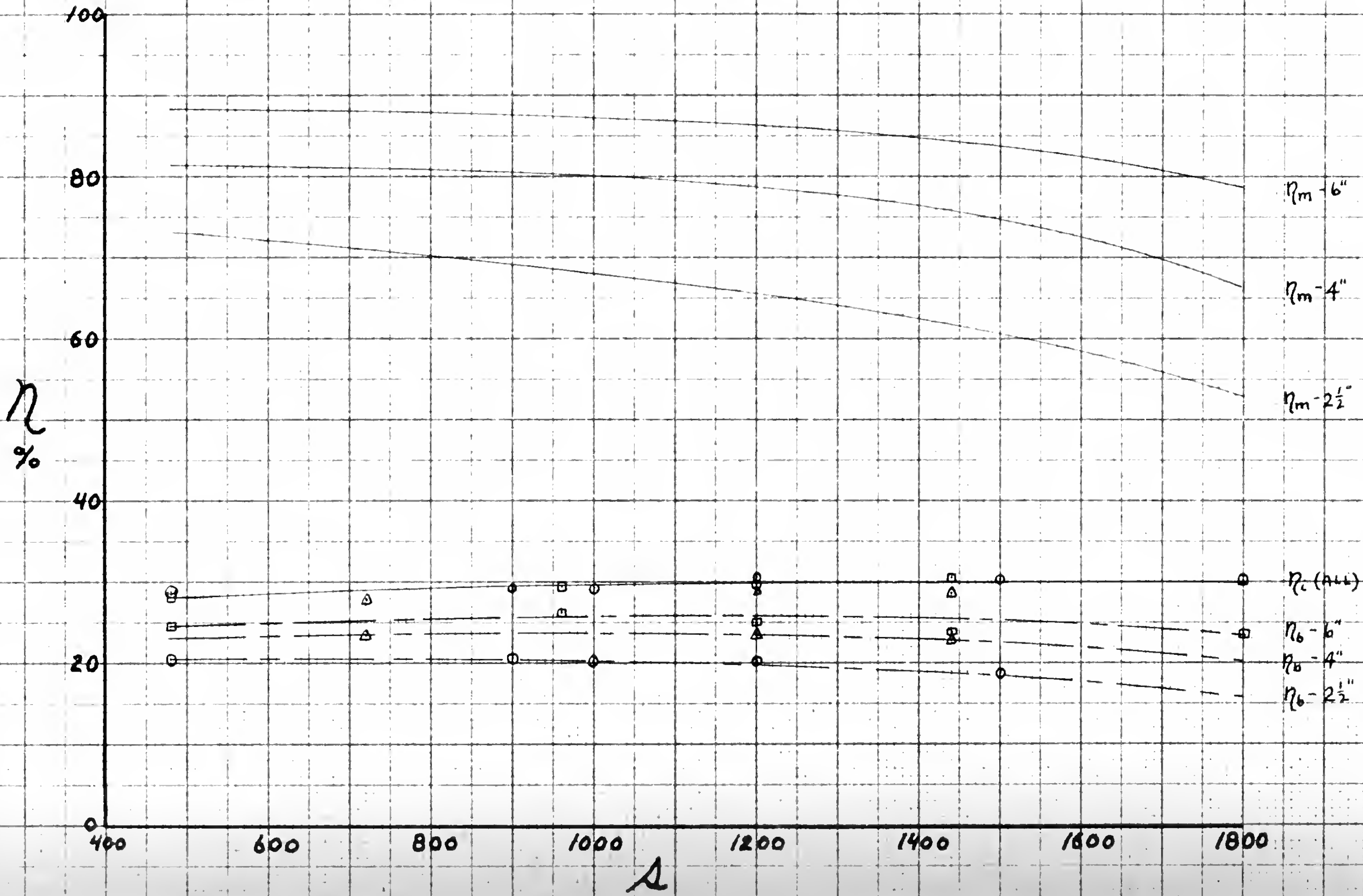


FIG. 8

FIG. 9

G.S. ENGINES

PRESSURE VOLUME DIAGRAMS

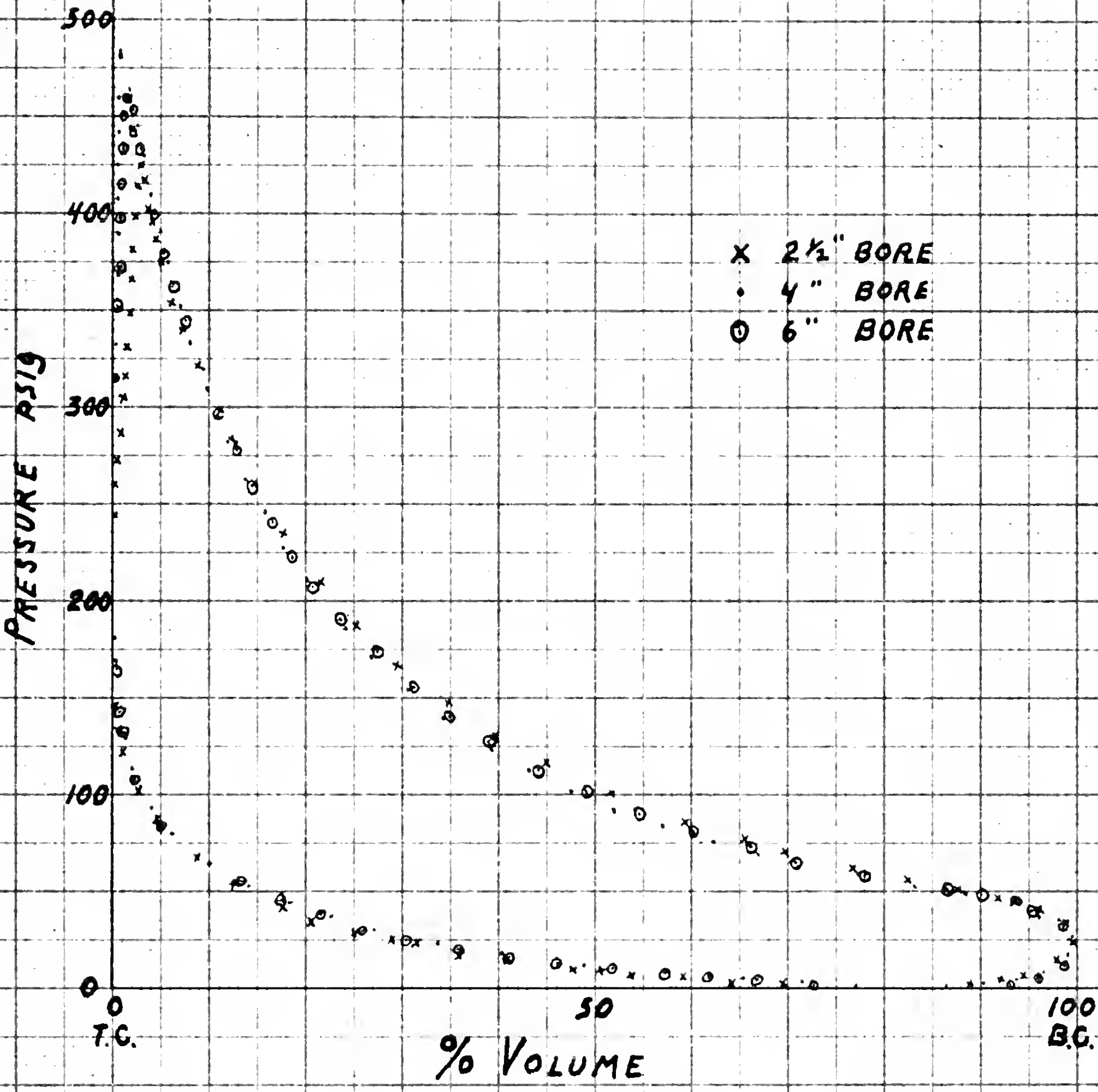
FROM INDICATOR CARDS

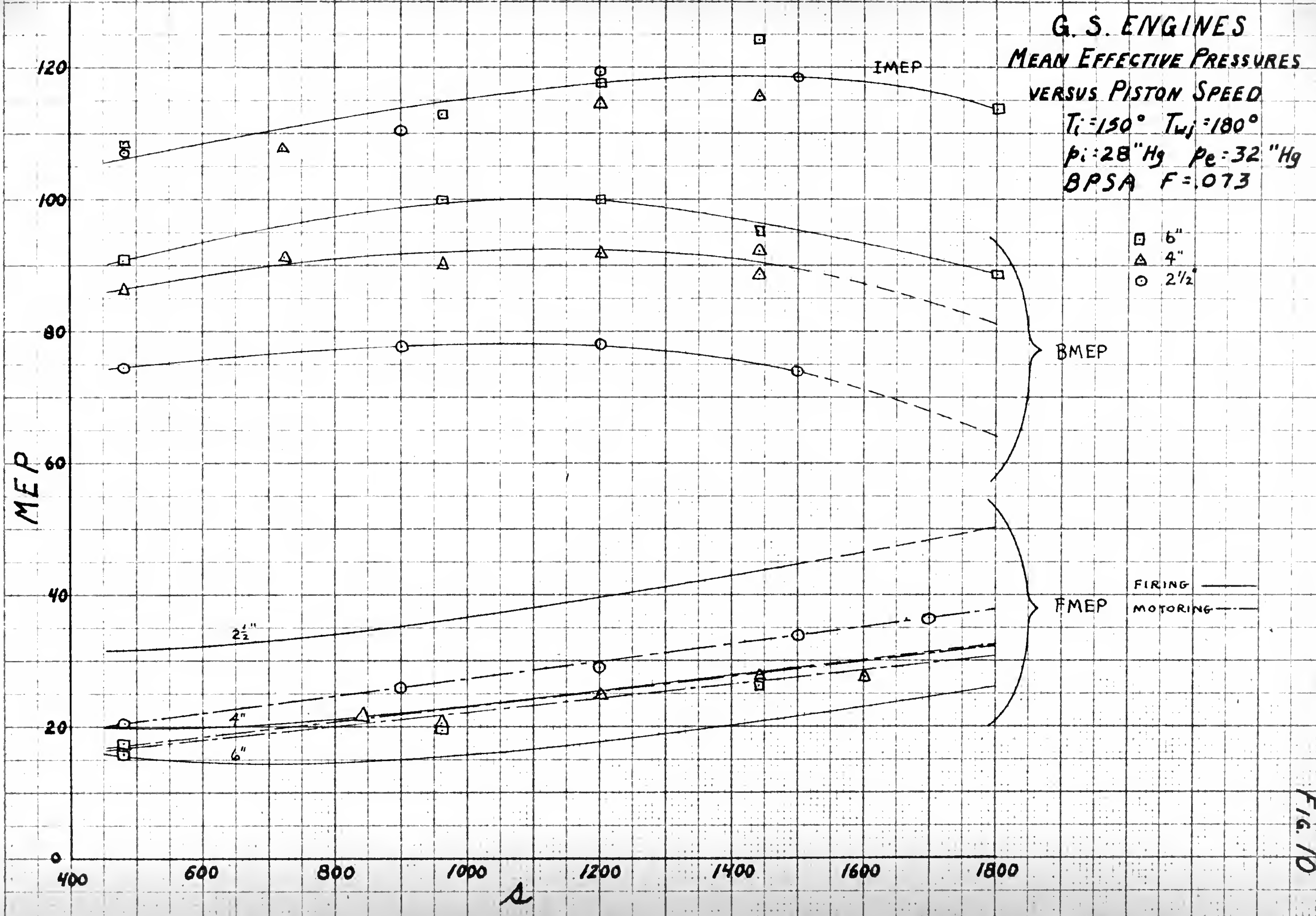
A = 1200

BPSA F = .073

T_i = 150 p_i = 28" Hg

T_{wj} = 180 p_e = 32" Hg





G. S. ENGINES
MEAN EFFECTIVE PRESSURES
VERSUS PISTON SPEED
$T_i = 150°$ $T_{wj} = 180°$
$p_i : 28$"Hg $p_e : 32$"Hg
BPSA F = .073
6"
4"
2 1/2"
IMEP
BMEP
FMEP
FIRING
MOTORING
2 1/2"
4"
6"
MEP
120
100
80
60
40
20
0
400
600
800
1000
1200
1400
1600
1800
s

FIG. 10

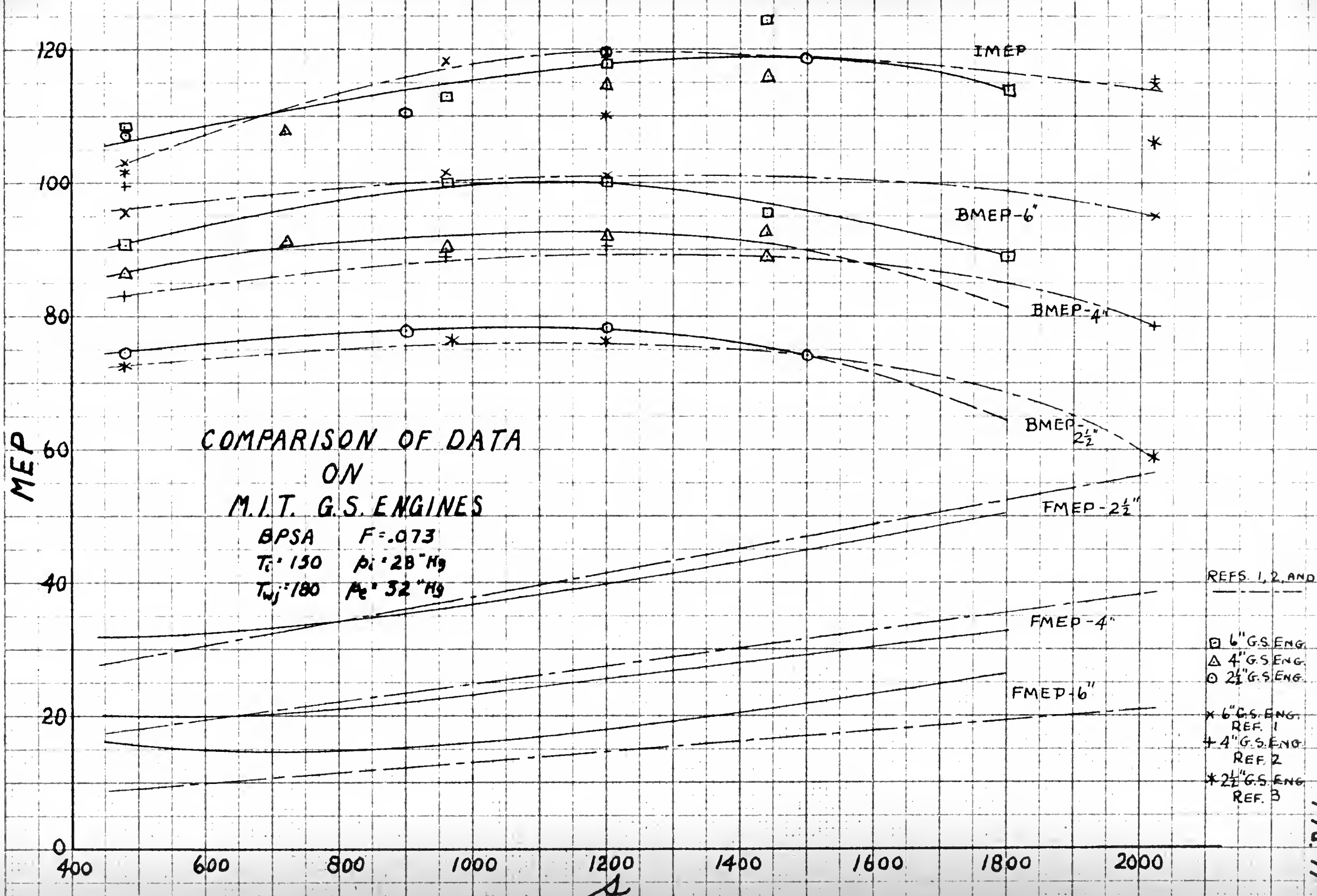
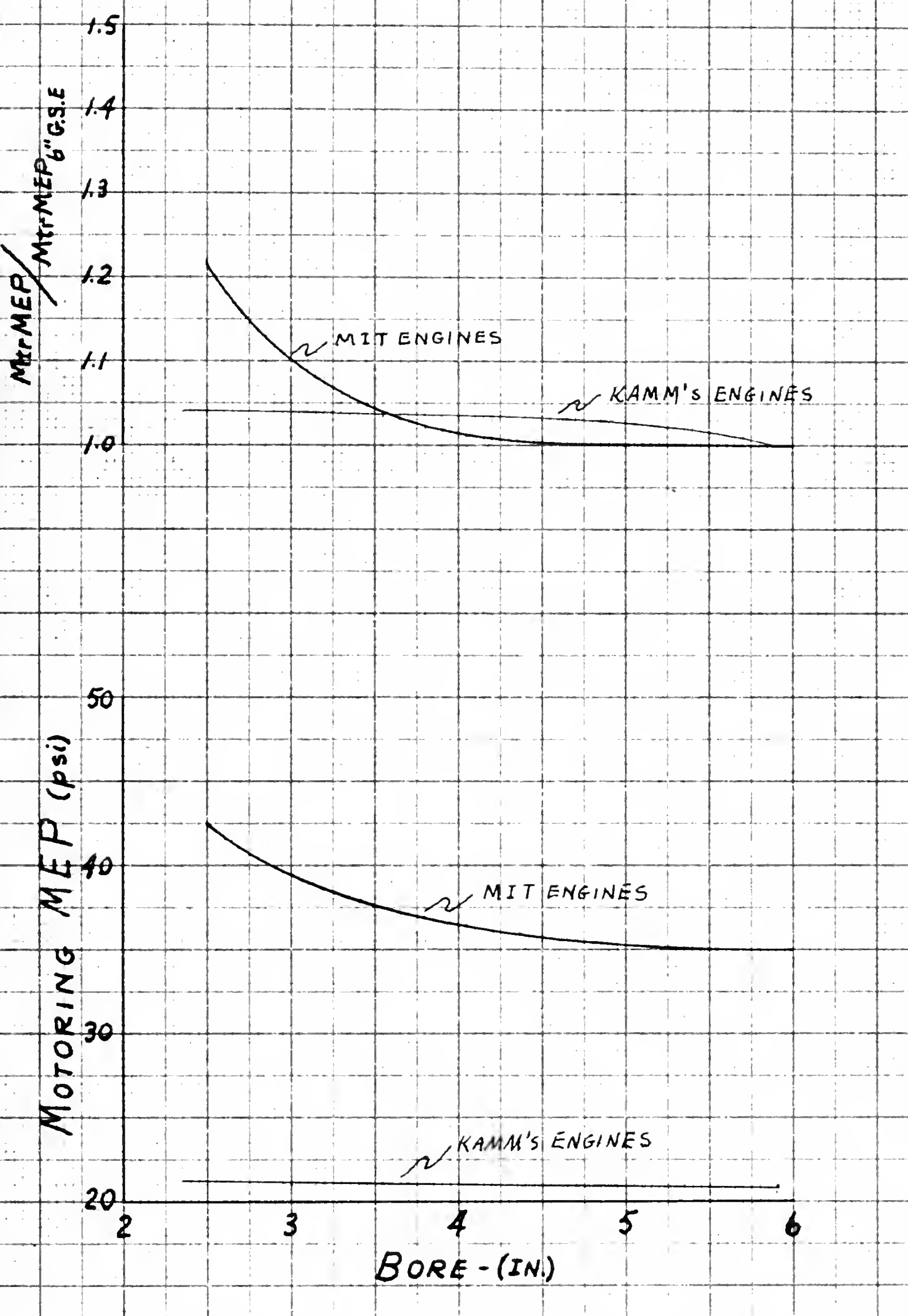

COMPARISON OF DATA
ON
M.I.T. G.S. ENGINES
BPSA F=.073
T_i=150 p_i=28"Hg
T_{wj}=180 p_e=32"Hg
IMEP
BMEP-6"
BMEP-4"
BMEP-2½"
FMEP-2½"
FMEP-4"
FMEP-6"
REFS. 1, 2, AND 3
6" G.S. ENG.
4" G.S. ENG.
2½" G.S. ENG.
6" G.S. ENG. REF. 1
4" G.S. ENG. REF. 2
2½" G.S. ENG. REF. 3
MEP
120
100
80
60
40
20
0
400
600
800
1000
1200
1400
1600
1800
2000
s

FIG. 11

Fig. 12

Comparison of Motoring Friction Data for G.S. Engines



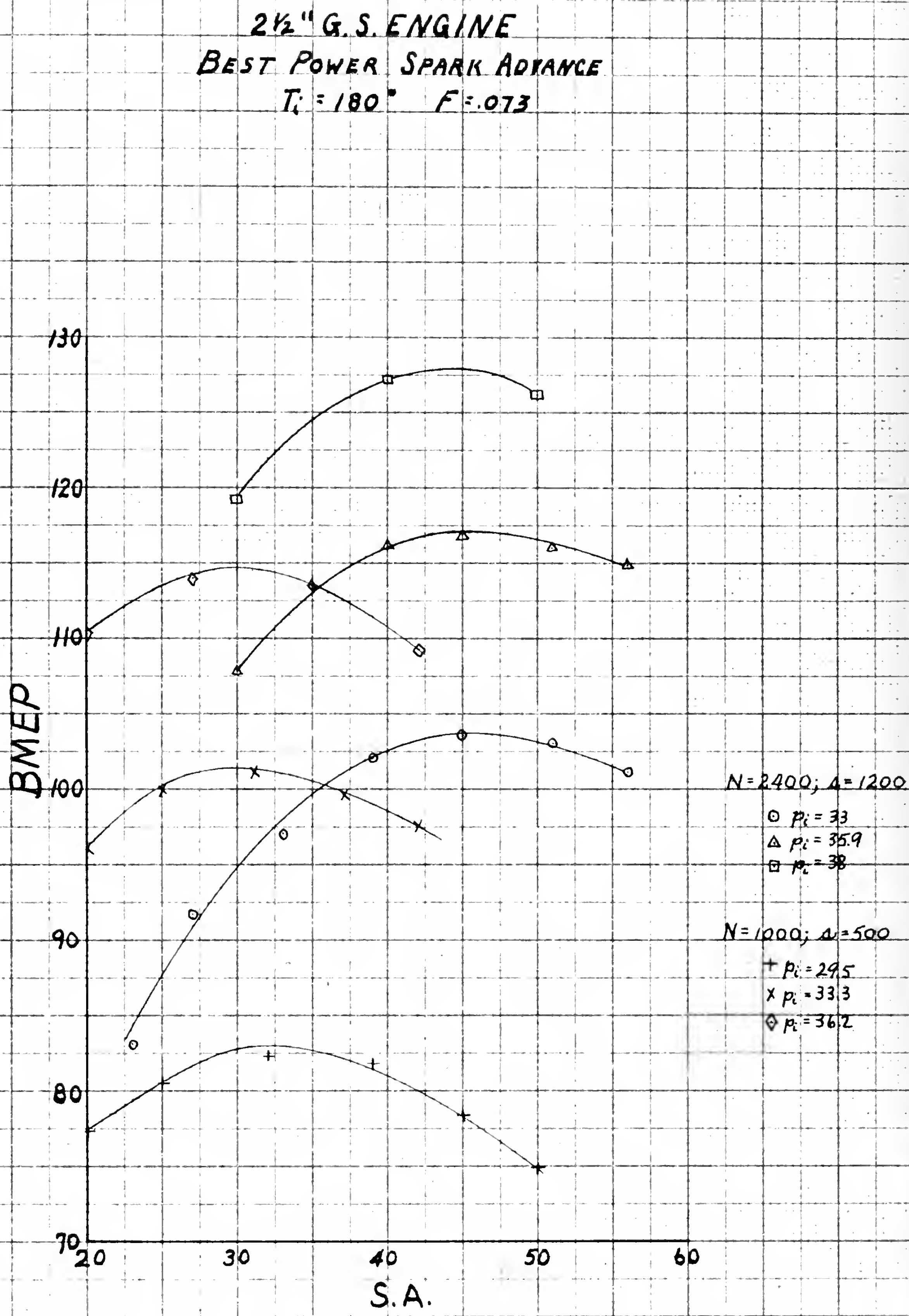

2½" G. S. ENGINE
BEST POWER SPARK ADVANCE
T_i = 180° F = .073
BMEP
S.A.
130
120
110
100
90
80
70
20
30
40
50
60
N = 2400; Δ = 1200
⊙ p_i = 33
△ p_i = 35.9
⊡ p_i = 38
N = 1000; Δ = 500
+ p_i = 29.5
× p_i = 33.3
◇ p_i = 36.2

Fig. 14

4" G.S.E.
BMEP versus S.A.
F = .073
T_i = 180

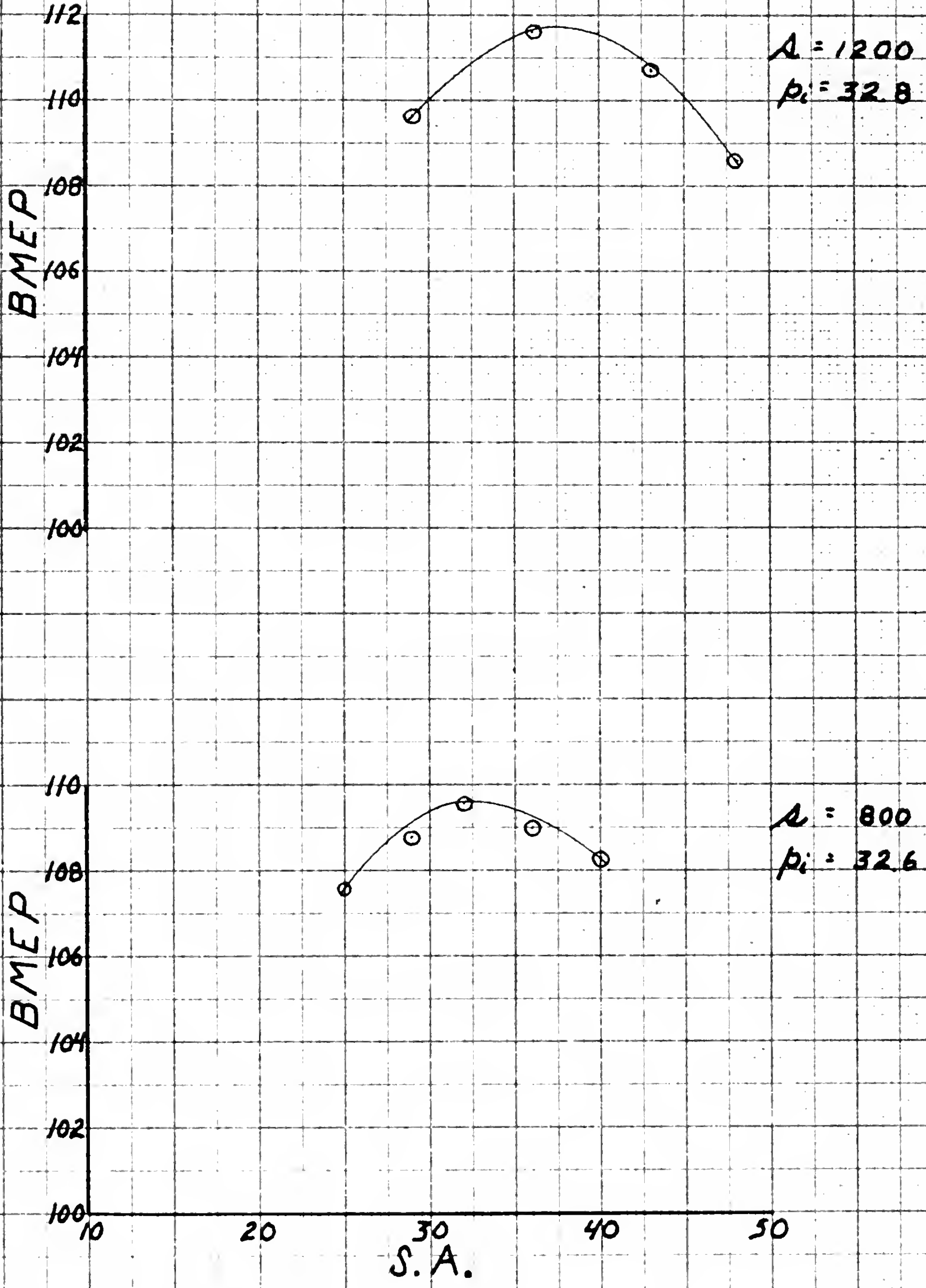


FIG. 15

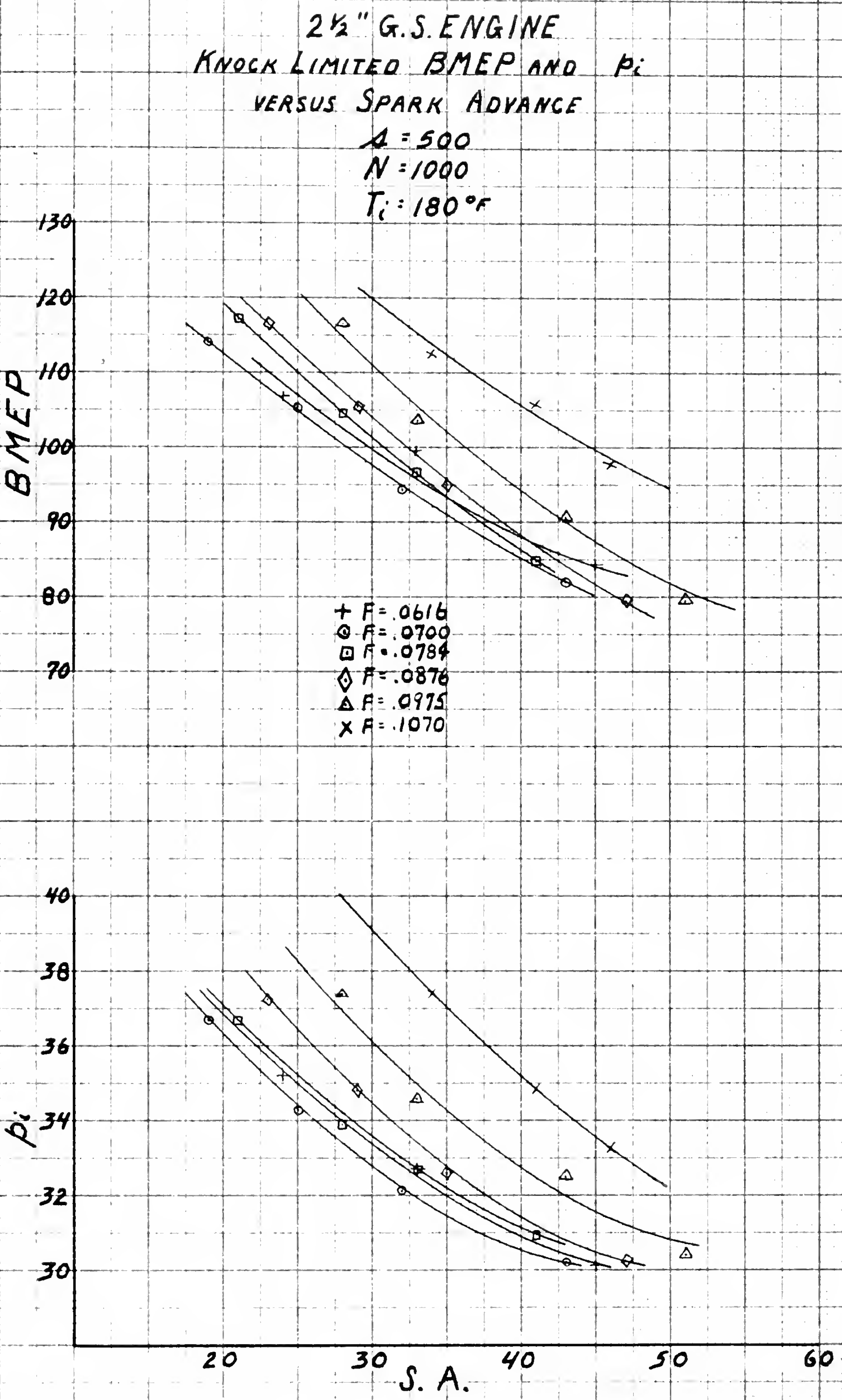

2½" G.S. ENGINE
KNOCK LIMITED BMEP AND p_i
VERSUS SPARK ADVANCE
Δ = 500
N = 1000
T_i = 180°F
BMEP
130
120
110
100
90
80
70
+ F = .0616
⊙ F = .0700
⊡ F = .0784
◇ F = .0876
△ F = .0975
X F = .1070
p_i
40
38
36
34
32
30
20
30
40
50
60
S. A.

FIG. 16

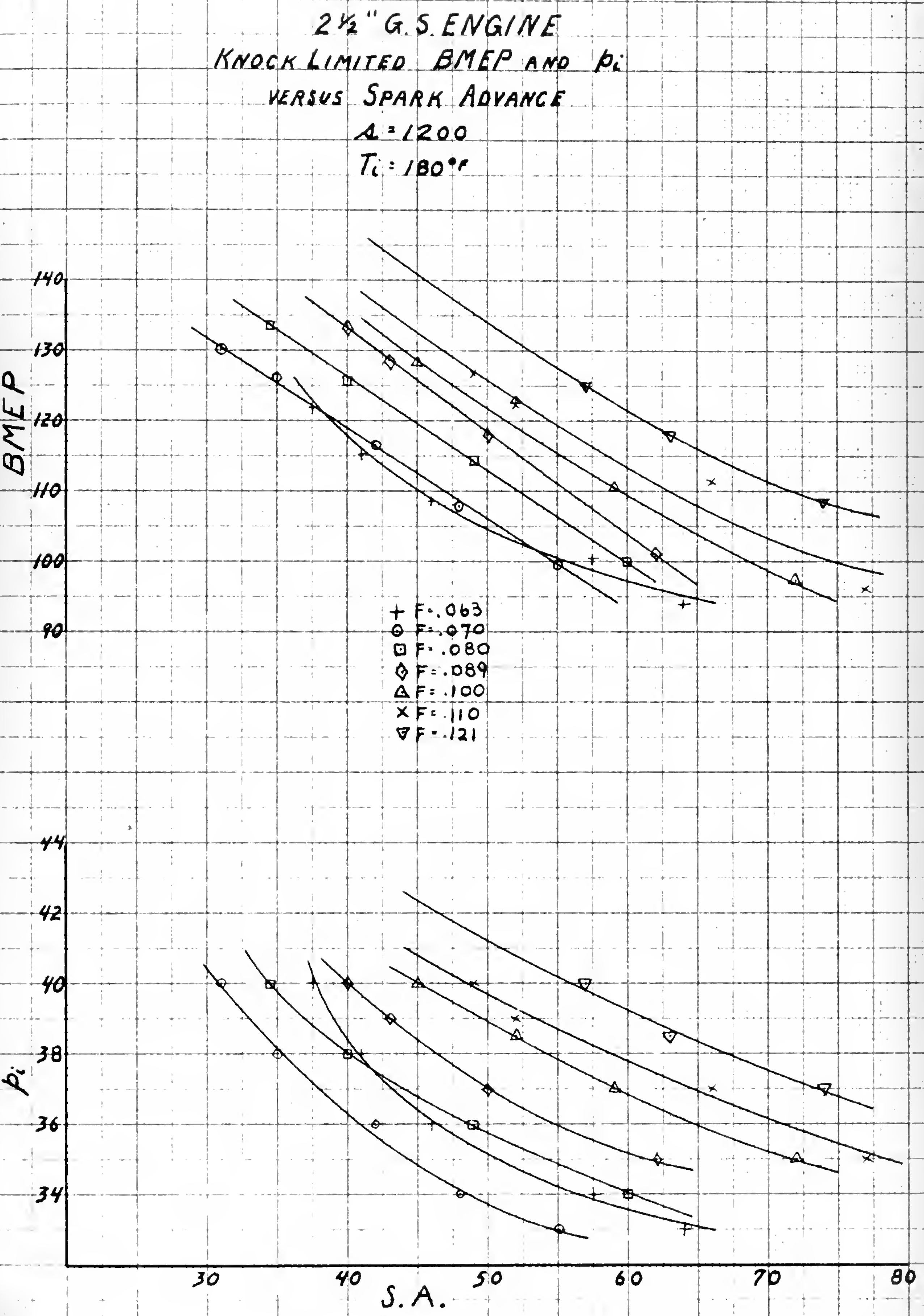
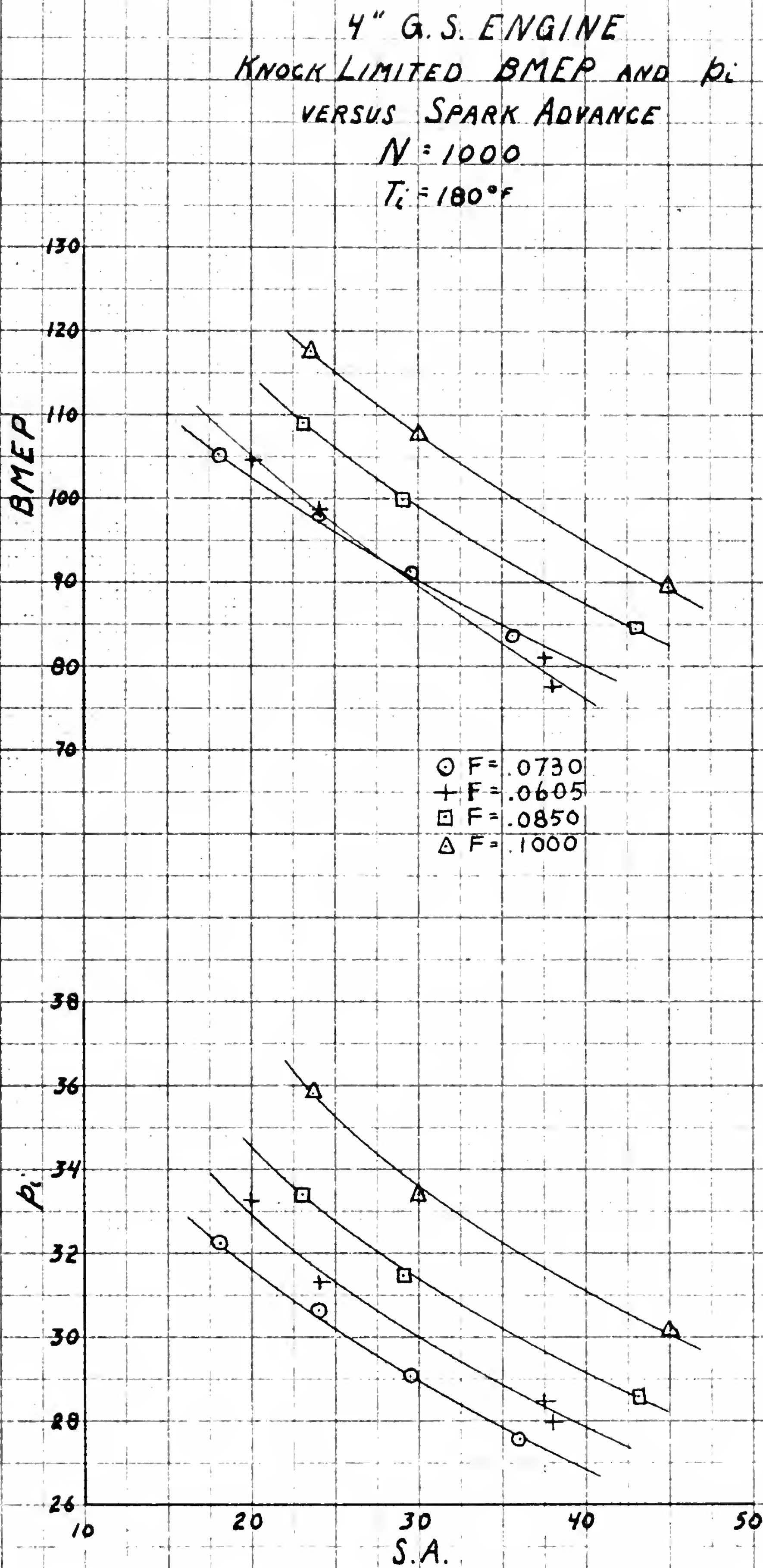

2½" G.S. ENGINE
KNOCK LIMITED BMEP AND p_i
VERSUS SPARK ADVANCE
N = 1200
T_i = 180°F
BMEP
140
130
120
110
100
90
+ F = .063
⊙ F = .070
⊡ F = .080
◊ F = .089
Δ F = .100
× F = .110
▽ F = .121
p_i
44
42
40
38
36
34
30
40
50
60
70
80
S.A.

FIG. 17



4" G. S. ENGINE
KNOCK LIMITED BMEP AND p_i
VERSUS SPARK ADVANCE
N = 1000
T_i = 180°F
BMEP
130
120
110
100
90
80
70
○ F = .0730
+ F = .0605
□ F = .0850
△ F = .1000
p_i
38
36
34
32
30
28
26
10
20
30
40
50
S.A.

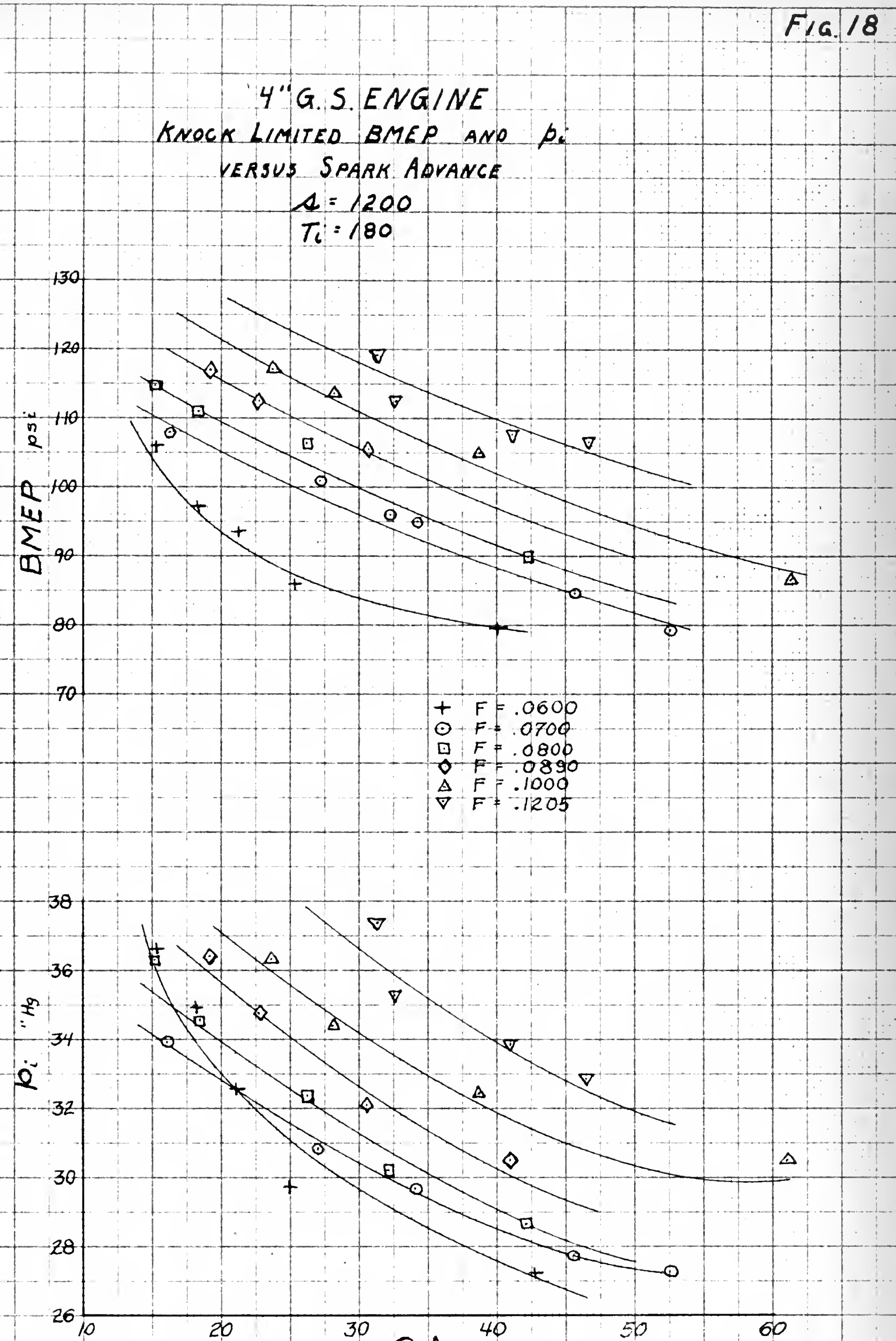
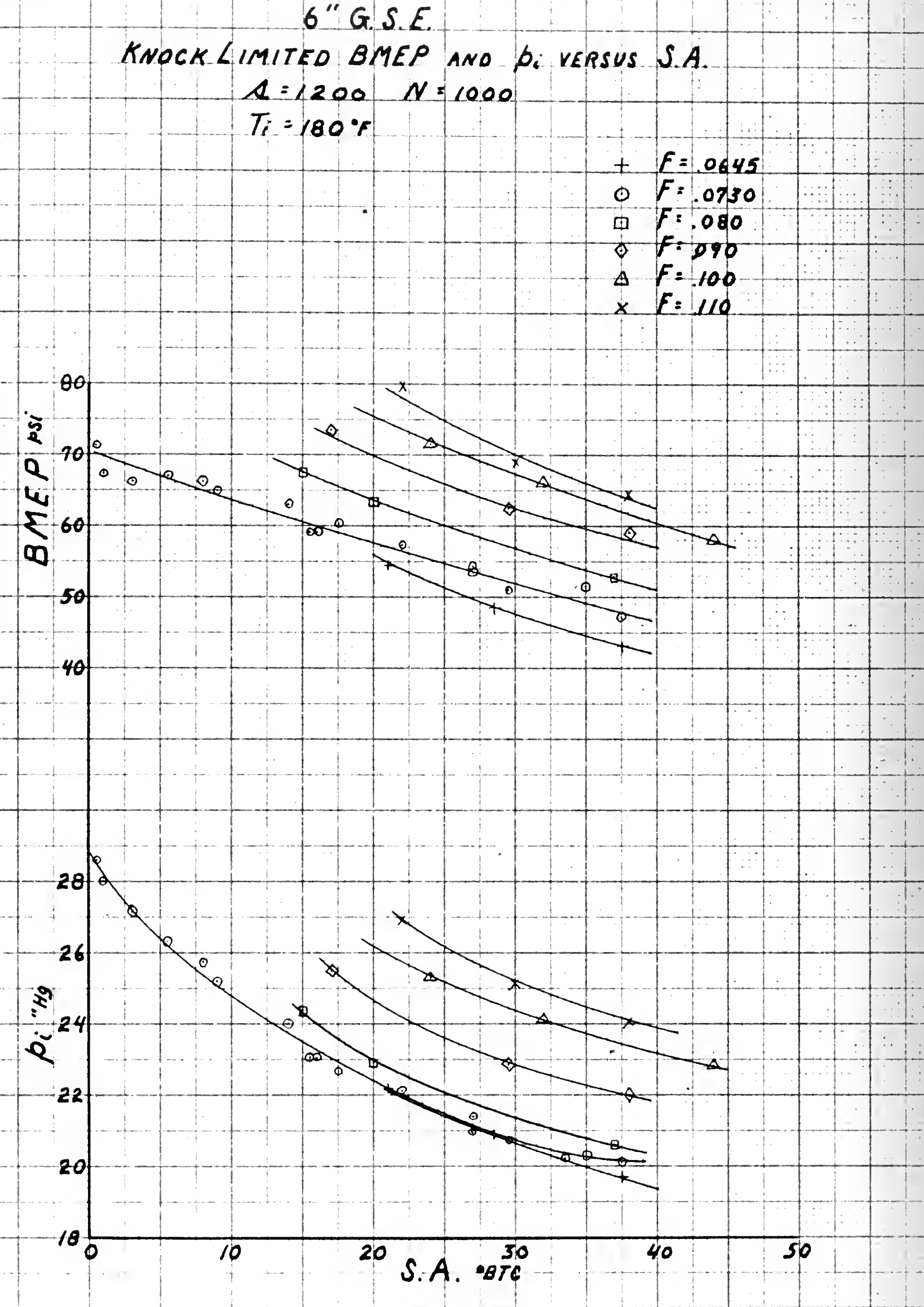


FIG. 18
4" G. S. ENGINE
KNOCK LIMITED BMEP AND p_i
VERSUS SPARK ADVANCE
A = 1200
T_i = 180
BMEP psi
130
120
110
100
90
80
70
F = .0600
F = .0700
F = .0800
F = .0890
F = .1000
F = .1205
p_i "Hg
38
36
34
32
30
28
26
10
20
30
40
50
60
S.A. °BTC


6" G.S.E.
KNOCK LIMITED BMEP AND p_i VERSUS S.A.
A = 1200 N = 1000
T_i = 180°F
+ F = .0645
⊙ F = .0730
□ F = .080
◇ F = .090
△ F = .100
× F = .110
BMEP psi
80
70
60
50
40
p_i "Hg
28
26
24
22
20
18
0
10
20
30
40
50
S.A. °BTC

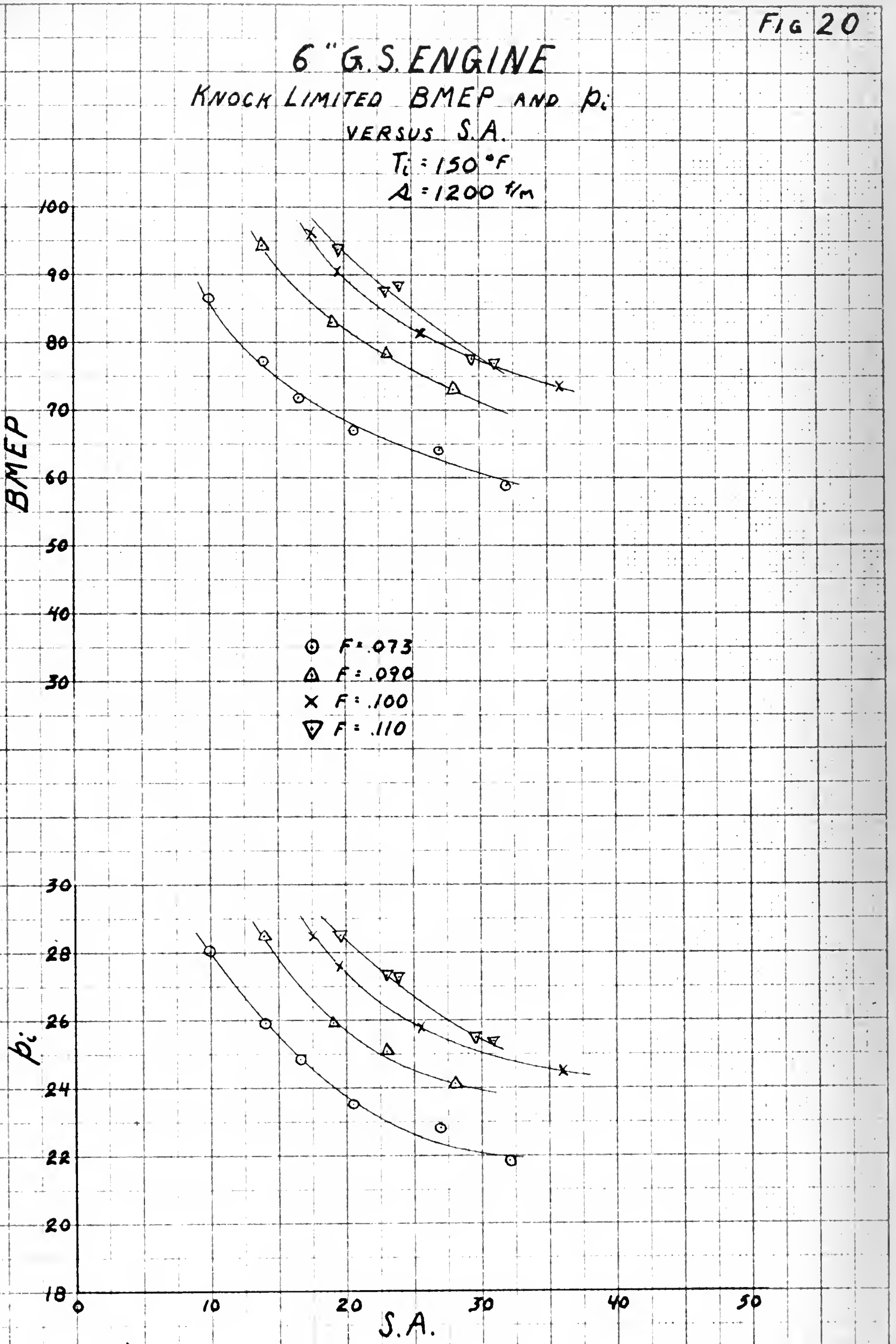


FIG 20
6" G.S. ENGINE
KNOCK LIMITED BMEP AND p_i
VERSUS S.A.
T_i = 150 °F
A = 1200 r/m
BMEP
p_i
S.A.
F = .073
F = .090
F = .100
F = .110
100
90
80
70
60
50
40
30
28
26
24
22
20
18
0
10
20
30
40
50

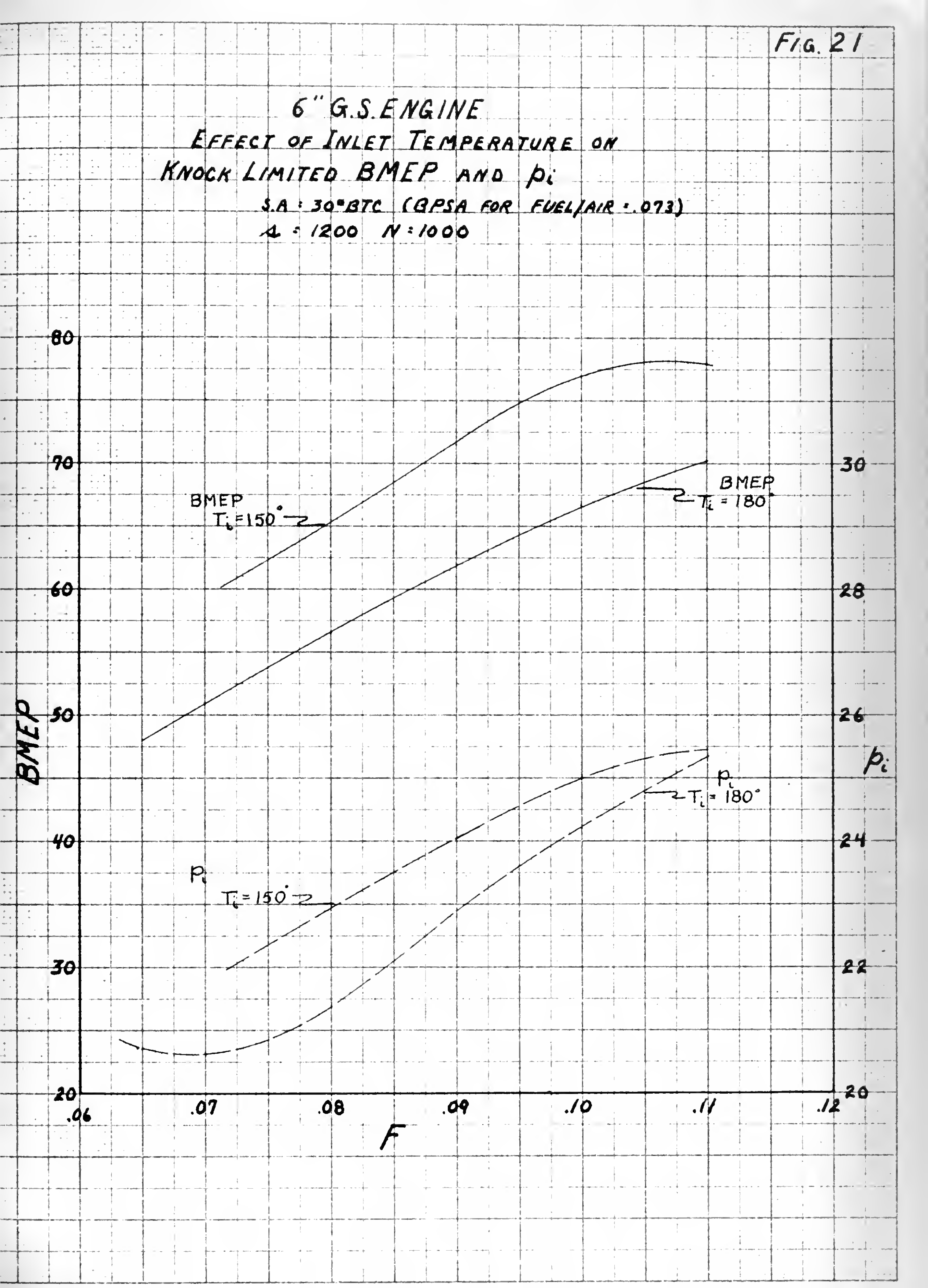
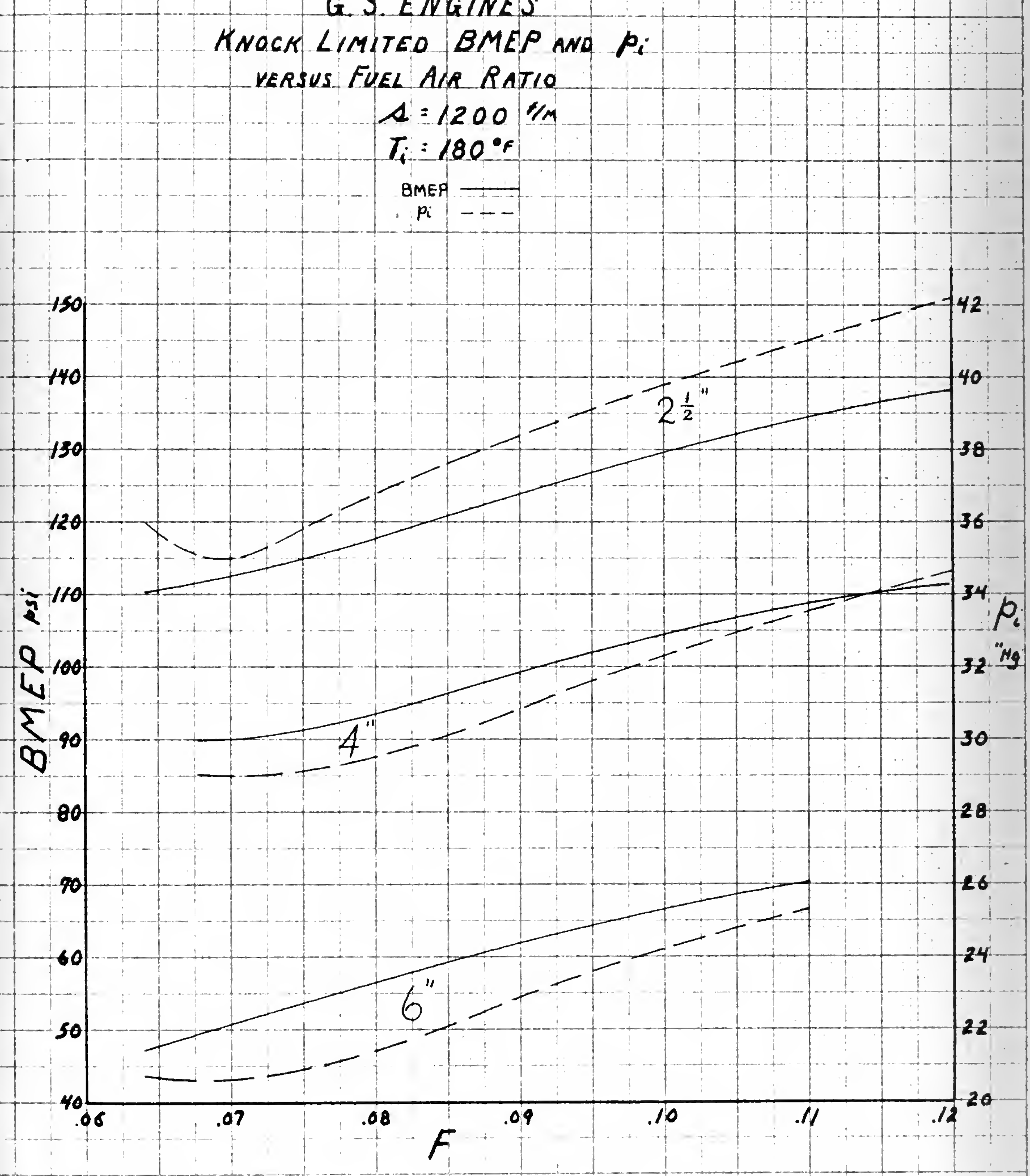


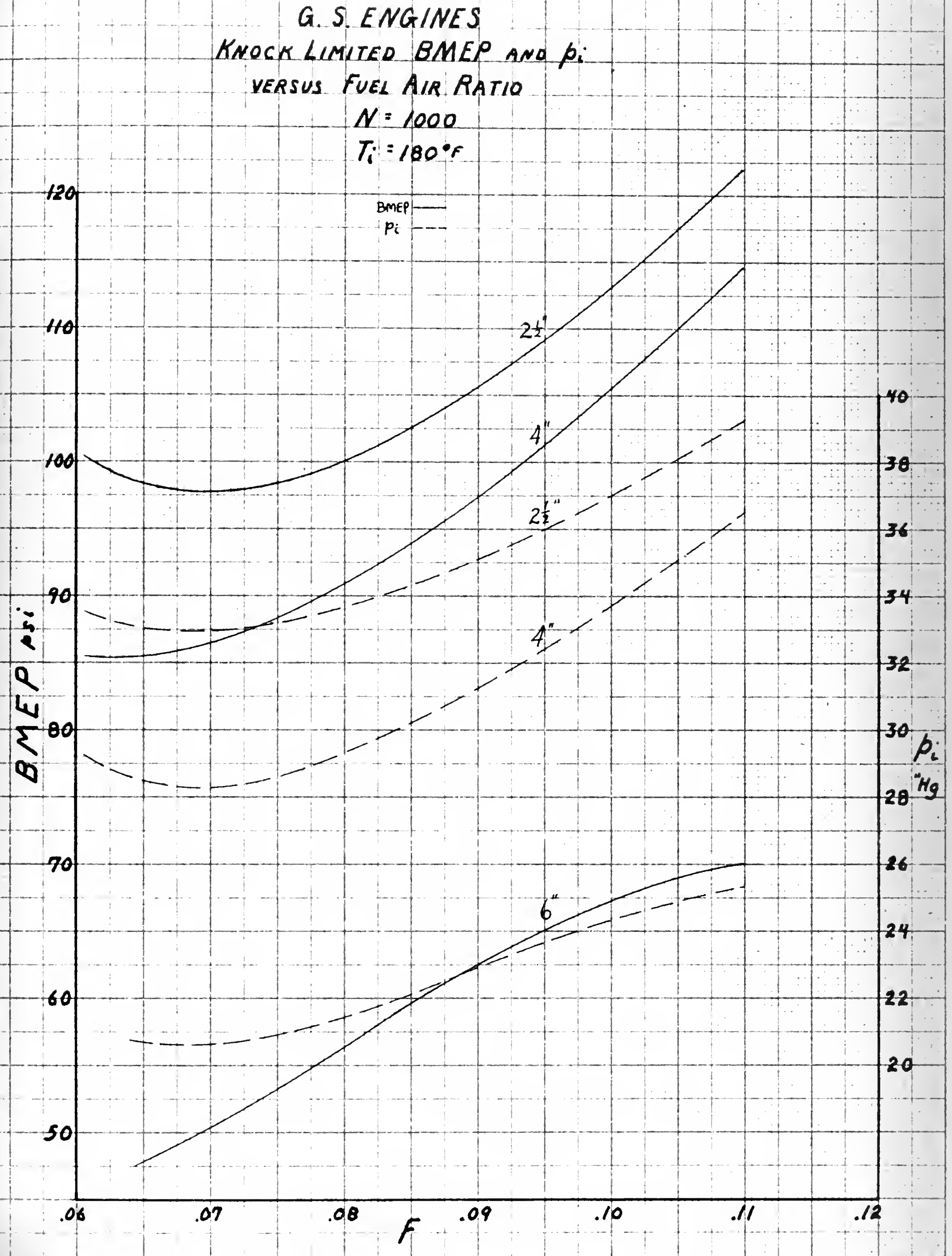
FIG. 21
6" G.S. ENGINE
EFFECT OF INLET TEMPERATURE ON
KNOCK LIMITED BMEP AND p_i
S.A = 30° BTC (BPSA FOR FUEL/AIR = .073)
A = 1200 N = 1000
BMEP T_i = 150°
BMEP T_i = 180°
p_i T_i = 150°
p_i T_i = 180°
BMEP
p_i
F
80
70
60
50
40
30
20
30
28
26
24
22
20
.06
.07
.08
.09
.10
.11
.12

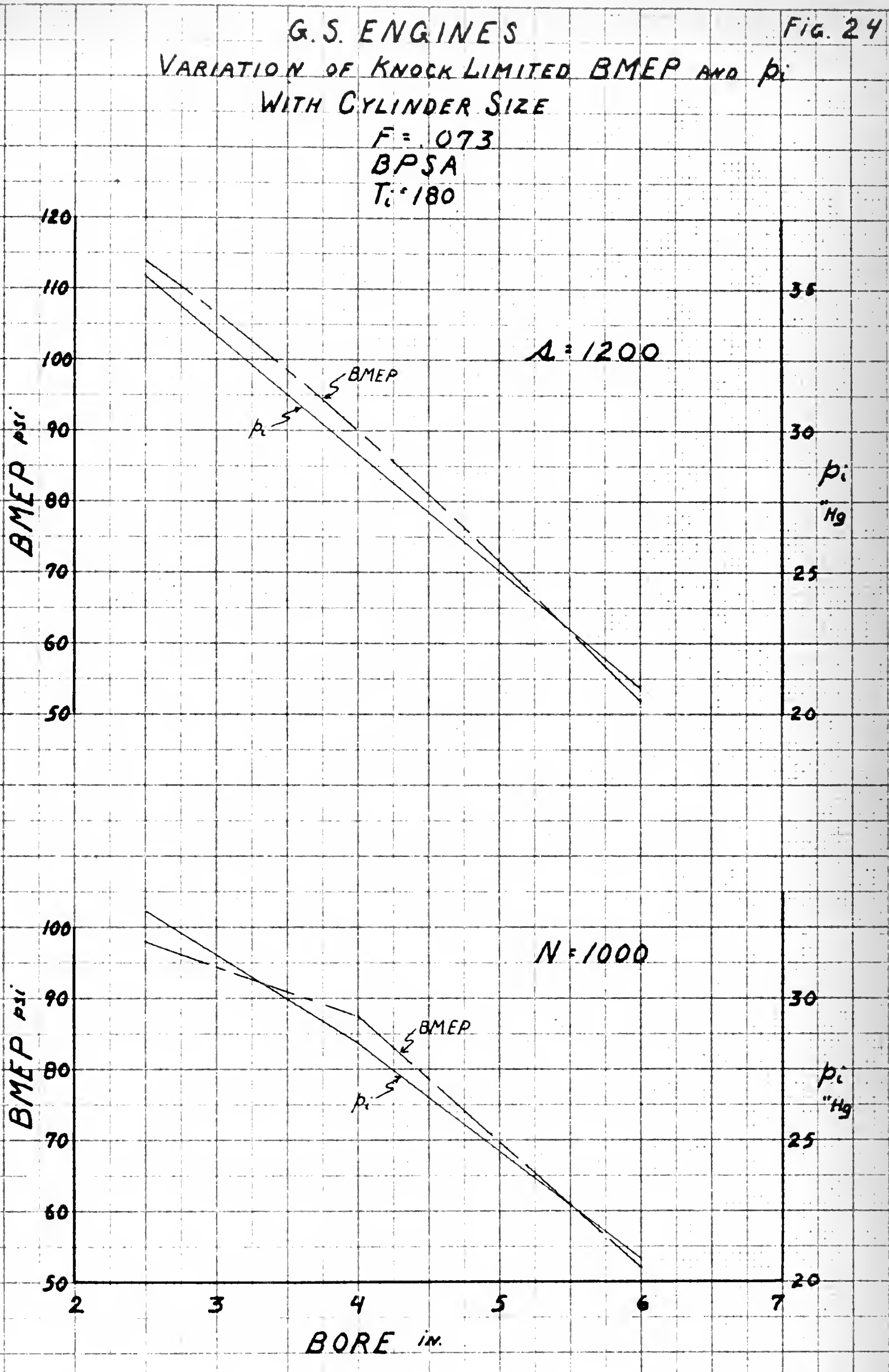
Fig. 22



G. S. ENGINES
KNOCK LIMITED BMEP AND p_i
VERSUS FUEL AIR RATIO
s = 1200 f/m
T_i = 180 °F
BMEP ———
p_i - - -
$2\frac{1}{2}$"
4"
6"
BMEP psi
p_i "Hg
F
150
140
130
120
110
100
90
80
70
60
50
40
42
40
38
36
34
32
30
28
26
24
22
20
.06
.07
.08
.09
.10
.11
.12

FIG. 23


G. S. ENGINES
KNOCK LIMITED BMEP AND p_i
VERSUS FUEL AIR RATIO
N = 1000
T_i = 180°F
BMEP
p_i
120
110
100
90
80
70
60
50
BMEP psi
40
38
36
34
32
30
28
26
24
22
20
p_i
"Hg
2½"
4"
2½"
4"
6"
.06
.07
.08
.09
.10
.11
.12
F



G. S. ENGINES
FIG. 24
VARIATION OF KNOCK LIMITED BMEP AND pi
WITH CYLINDER SIZE
F = .073
BPSA
Ti = 180
A = 1200
BMEP
pi
BMEP psi
pi "Hg
120
110
100
90
80
70
60
50
35
30
25
20
N = 1000
100
90
80
70
60
50
2
3
4
5
6
7
BORE IN.

FIG. 25

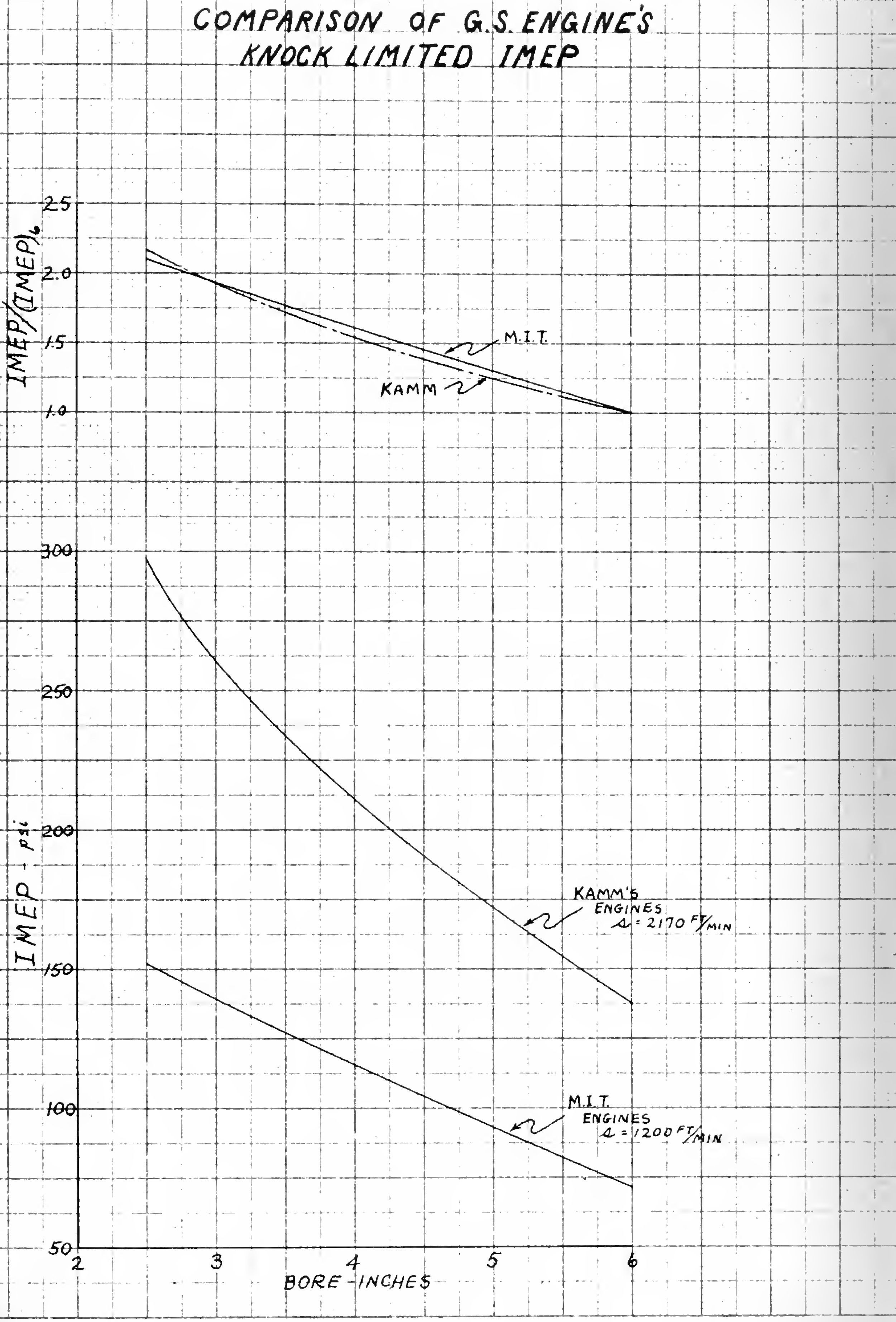
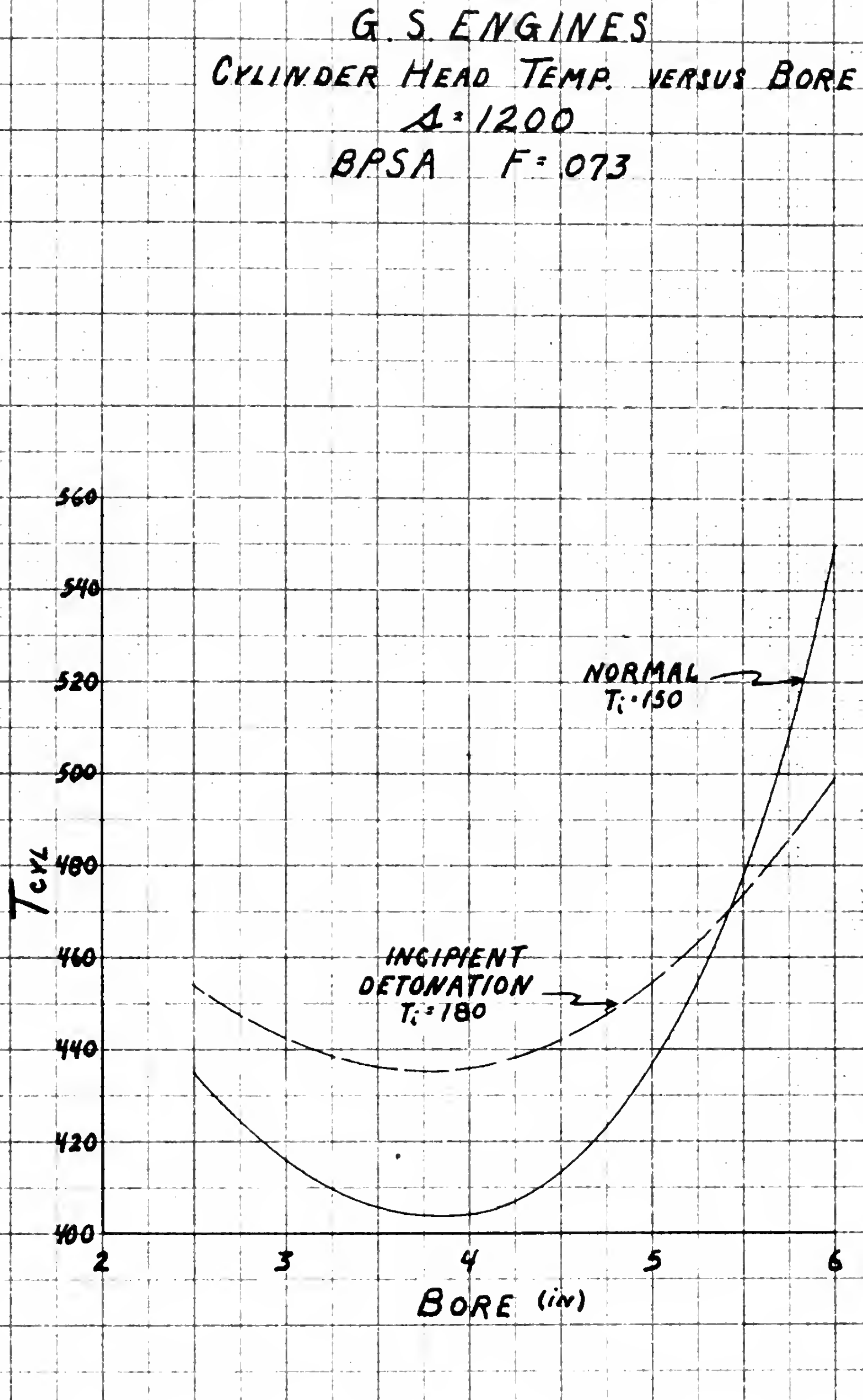

COMPARISON OF G.S. ENGINE'S
KNOCK LIMITED IMEP
IMEP/(IMEP)6
2.5
2.0
1.5
1.0
M.I.T.
KAMM
300
250
200
150
100
50
IMEP - psi
KAMM'S
ENGINES
s = 2170 FT/MIN
M.I.T.
ENGINES
s = 1200 FT/MIN
2
3
4
5
6
BORE - INCHES

FIG 26

G.S. ENGINES
VARIATION OF CYLINDER HEAD TEMP.
WITH FUEL-AIR RATIO
AT INCIPIENT DETONATION
$T_i = 180°F$
$\Delta = 1200$
BPSA AT $F = .073$

FIG. 27


G.S. ENGINES
CYLINDER HEAD TEMP. VERSUS BORE
Δ = 1200
BPSA F = .073
NORMAL
T_i = 150
INCIPIENT
DETONATION
T_i = 180
T_{CYL}
560
540
520
500
480
460
440
420
400
2
3
4
5
6
BORE (in)

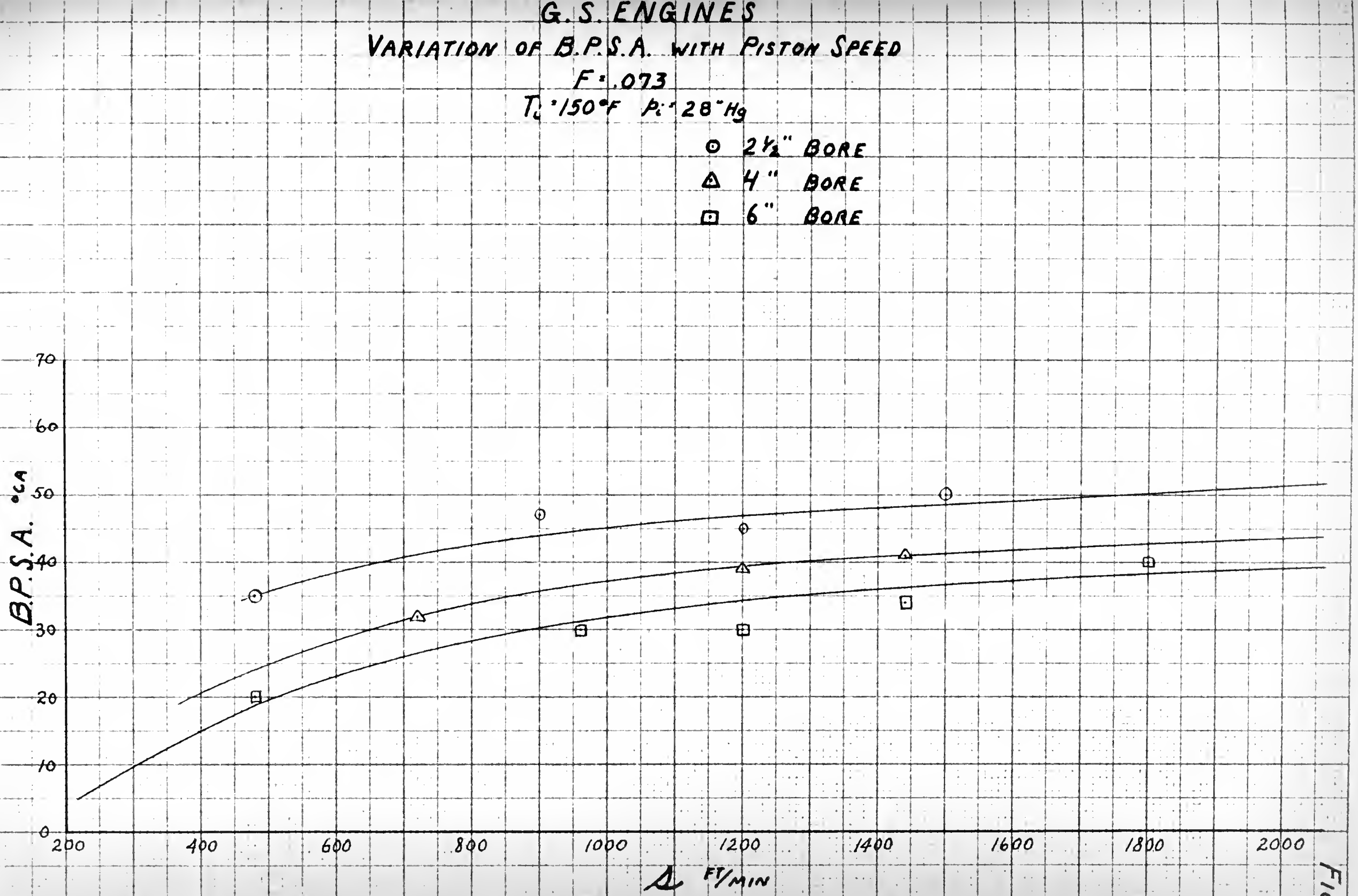

G. S. ENGINES
VARIATION OF B.P.S.A. WITH PISTON SPEED
F = .073
T_i = 150°F P_i = 28" Hg
⊙ 2½" BORE
△ 4" BORE
⊡ 6" BORE
B.P.S.A. °CA
70
60
50
40
30
20
10
0
200
400
600
800
1000
1200
1400
1600
1800
2000
S FT/MIN

FIG. 28

G.S. ENGINES

VARIATION OF FLAME SPEED WITH REYNOLD'S NUMBER
FOR CONSTANT PISTON SPEEDS

$T_i = 150°F$ $p_i = 28''Hg$

BPSA   F = .073

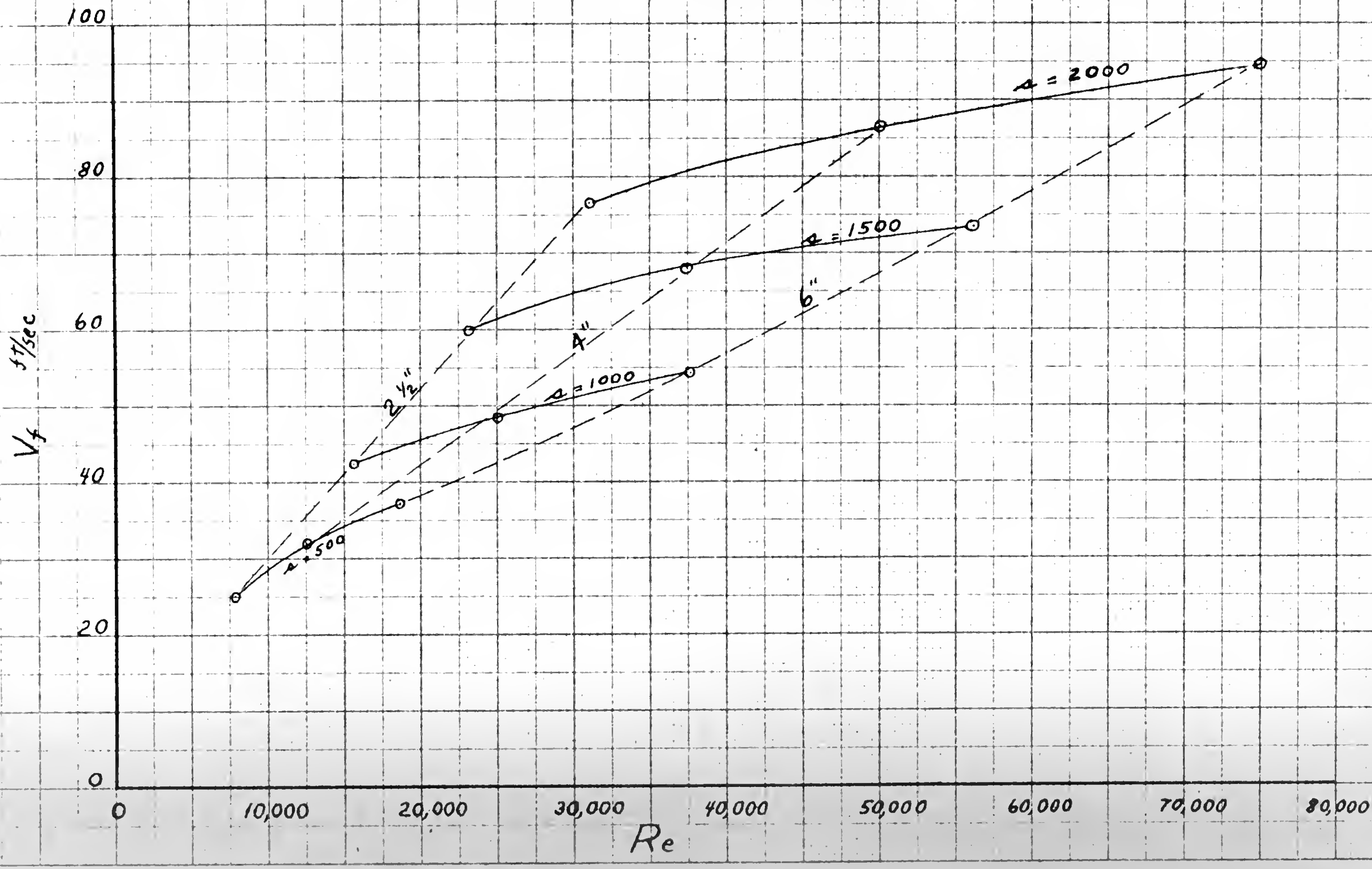


FIG. 29

G.S. ENGINES

VARIATION OF FLAME SPEED WITH PISTON SPEED FOR CONSTANT REYNOLD'S NUMBERS

T_i : 150°F p_i : 28" Hg

BPSA F : .073

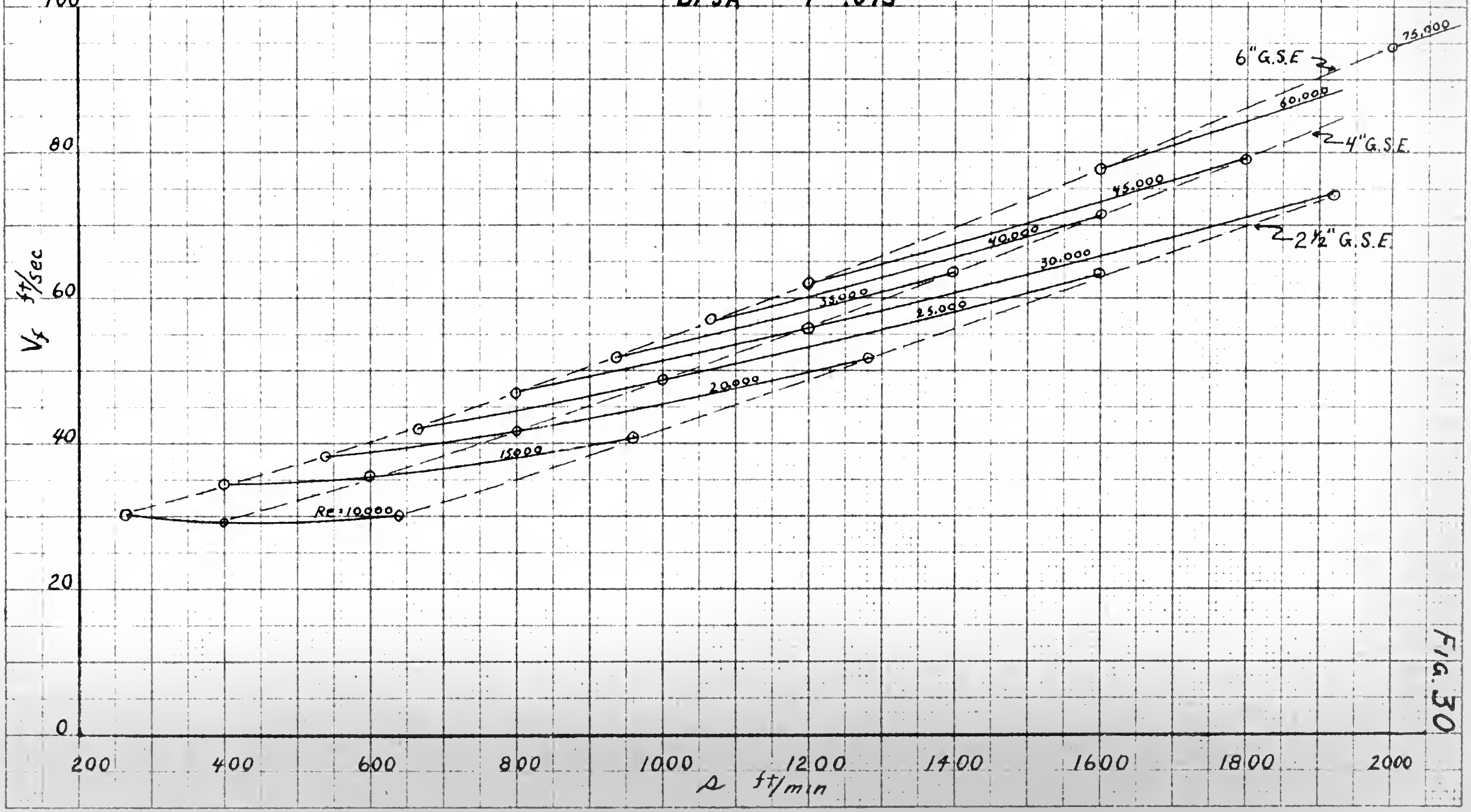
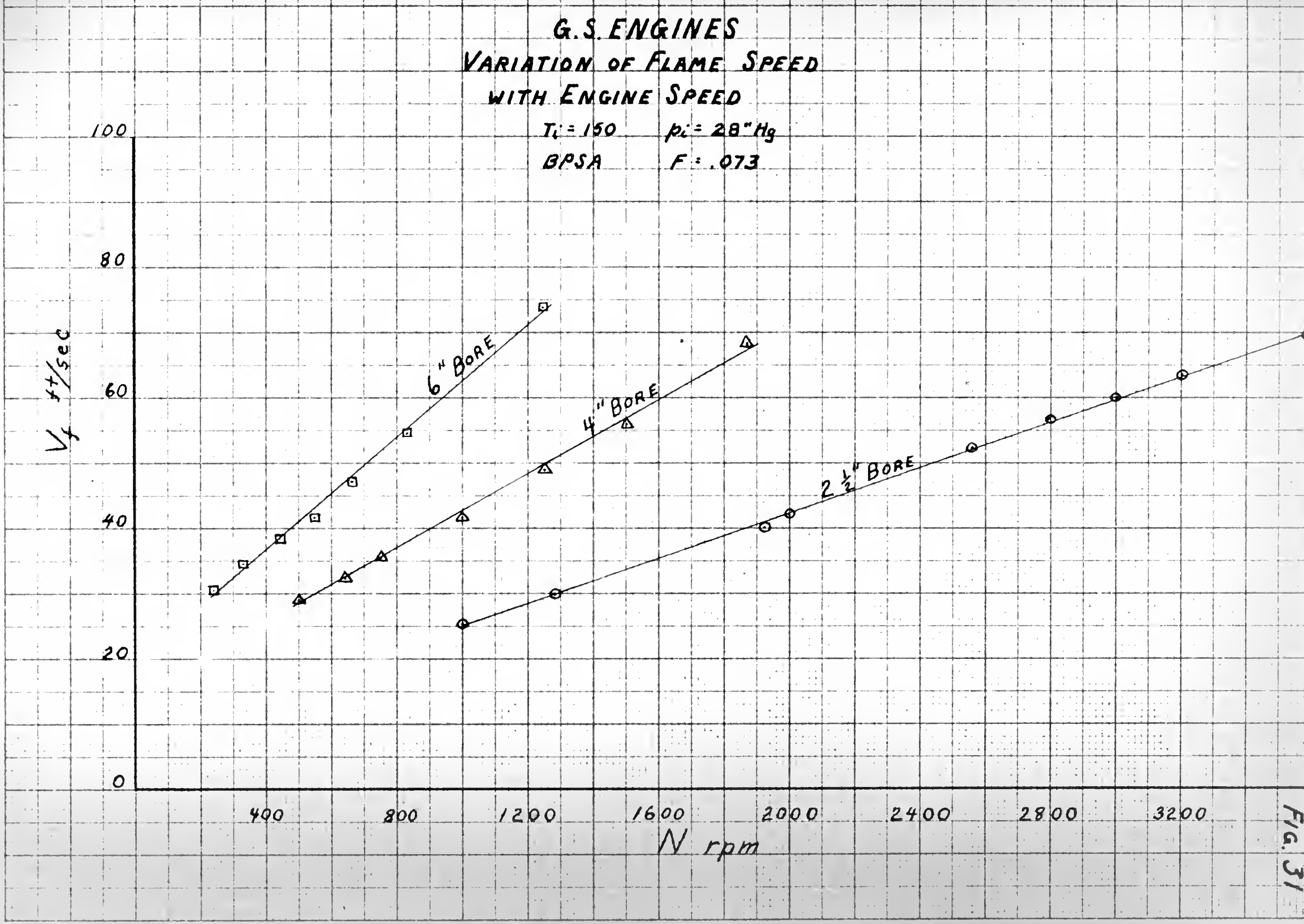


FIG. 30



G.S. ENGINES
VARIATION OF FLAME SPEED
WITH ENGINE SPEED
$T_i = 150$ $p_i = 28''$ Hg
BPSA $F = .073$
6" BORE
4" BORE
2 ½" BORE
V_f ft/sec
100
80
60
40
20
0
400
800
1200
1600
2000
2400
2800
3200
N rpm

FIG. 31

TABLE I

Uncorrected Data

2 1/2" M. I. T. G. S. Engine

Fuel 100 Octane

10 January 1950 (orifice diameter 0.413)

RPM	s	Spark Advance BTC	Roto-meter	$\dot{M}_f$	Δp Orifice	$\dot{M}_a$	F	Dyn. Scale	BMEP	T_i	T_{cyl}	T_{wj}	T_a	T_{oil}	P_i	P_e	P_a
*2400	1200	30	7.25	.000704	10.55	.00887	.0794	20.6	74	150	392	180	82	150	28	32	30.2
"	"	40	7.25	.000704	10.50	.00884	.0796	22.6	81.1	149	415	181	"	151	"	"	"
"	"	50	7.25	.000704	10.45	.00883	.0797	22.2	79.7	150	418	180	"	150	"	"	"

Sample Calculations

$$\dot{M}_a = .1145\ D_2^{\,2} K Y_1 \sqrt{\frac{P_a}{T_a} Gyh}$$ (See Air Flow Measurements Appendix E)

Barometer	P_a	T_a	K approx.	Y approx.	Gy	h	$\dot{M}_a$	F	BMEP
767.9	30.2	82	.99	.6	1	Δp orifice	$.00278\sqrt{h}$	Mf/Ma	3.59 (Dyn-Scale)

h	10.55	10.50	10.5
$\dot{M}_a$	.00887	.00884	.00883
F	.0794	.0796	.0797
Dyn-Scale	20.6	22.6	22.2
BMEP	74	81.1	79.7

Corrected Data

2 1/2" M.I.T.G.S. Engine

Fuel 100 Octane

10 January (orifice diam. 0.413)

RPM	s	Spark Advance	Roto-meter	$\dot{M}_f$	Δp orifice	$\dot{M}_a$	F	Dyn-Scale	BMEP	T_i	T_{cyl}	T_{wj}	T_a	T_{oil}	P_i	P_e	P_a
2400	1200	30	7.25	.000704	10.55	.00893	.0790	20.6	74	150	392	180	82	150	28	32	30.2
"	"	40	7.25	"	10.50	.00889	.0792	22.6	81	149	415	181	"	151	"	"	"
"	"	50	7.25	"	10.45	.00888	.0793	22.2	79.7	150	418	180	"	150	"	"	"

$\dot{M}_a(corr)/\dot{M}_a = 1.006$ (See Fig. E-1)

$\dot{M}_a$	.00887	.00884	.00883
$\dot{M}_a(corr)$	.00893	.00889	.00888
F	.0790	.0792	.0793

TABLE II
2 1/2" M. I. T. C. R. ENGINE
Fuel 100 octane

5 January 1950 (orifice diameter 0.413")

RPM	s	Spark Advance °BTC	Roto-meter	$\dot{m}_f$	Δp orifice	$\dot{m}_a$	F	Dyn. Scale	BMEP	T_i	T_{cyl}	T_{wj}	T_a	T_{oil}	P_i	P_e	P_a
960	480	50	3.85	.000200	1.55	.00348	.0575	20.0	71.8	150	320	176	81	150	28.0	32.0	30.1
"	"	40	3.85	.000200	"	"	.0575	20.2	72.5	"	310	180	"	151	"	32.0	"
"	"	30	3.80	.000195	"	"	.0561	19.5	70.0	"	"	181	"	150	"	32.1	"
"	"	20	3.85	.000200	"	"	.0575	16.7	60.0	"	"	181	"	150	"	32.0	"
"	"	35	"	"	"	"	"	20.2	72.5	"	315	182	"	149	"	"	"
"	"	45	"	"	"	"	"	20.35	73.1	"	320	182	"	149	"	"	"
960	480	45	4.1	.000228	1.52	.00344	.0672	20.45	73.5	150	330	181	81	150	28.0	32.0	30.1
"	"	40	"	"	"	"	"	21.15	76.0	"	330	"	"	149	"	"	"
"	"	30	"	"	"	"	"	21.7	78.0	"	320	"	"	150	"	"	"
"	"	20	4.05	.000223	"	"	.0649	20.1	72.2	"	318	178	80	150	"	"	"
"	"	35	4.1	.000228	1.53	.00345	.0661	21.5	77.2	"	325	180	"	149	"	"	"
"	"	50	4.05	.000223	1.50	.00342	.0653	20.1	72.2	"	340	181	"	149	"	"	"
6 January 1950 (orifice diameter 0.413")																	
960	480	35	4.4	.000264	1.52	.00344	.0766	21.05	75.7	150	320	182	82.5	150	28.0	32.0	30.0
"	"	30	"	"	"	"	"	21.05	75.7	"	318	182	82.5	151	"	"	"
"	"	20	"	"	"	"	"	20.5	73.6	"	312	181	83	150	"	"	"
"	"	40	"	"	"	"	"	20.4	73.3	149	325	"	"	150	"	"	"
"	"	50	"	"	1.50	.00342	.0771	19.65	70.5	149	330	"	"	151	"	"	"
960	480	50	3.85	.000200	1.53	.00345	.0580	19.95	71.6	150	320	181	83	150	28.0	32.0	30.0
960	480	50	4.75	.000310	1.50	.00342	.0906	18.75	67.2	150	315	180	83	150	28.0	32.0	30.0
"	"	40	4.80	.000316	1.50	.00342	.0924	20.1	72.2	151	300	179	"	"	"	"	"
"	"	30	"	"	1.51	.003435	.0920	20.4	73.3	151	"	179	"	"	"	"	"
"	"	35	"	"	1.51	.003435	.0920	20.5	73.6	150	"	180	"	151	"	"	"
"	"	20	"	"	1.53	.00345	.0916	19.4	69.6	149	299	180	"	150	"	"	"

TABLE III
2 1/2" M. I. T. C. S. Engine
Fuel 100 Octane

10 January 1950 (orifice diameter 0.413")

RPM	s	Spark Advance BTC	Roto-meter	$\dot{M}_f$	Δp orifice	$\dot{M}_a$	F	Dyn. Scale	BMEP	T_1	T_{cyl}	T_{wj}	T_a	T_{oil}	P_1	P_e	P_a
2400	1200	35	6.35	.000552	10.55	.00894	.0617	17.7	63.5	152	400	180	81	150	28.0	32.0	30.2
"	"	30	"	"	10.75	.00902	.0612	15.1	54.2	150	390	"	"	"	"	"	"
"	"	20	"	"	10.75	.00902	.0612	11.0	39.5	150	365	"	"	"	"	"	"
"	"	40	"	"	10.70	.00899	.0615	19.1	66.6	149	390	"	"	"	"	"	""
"	"	50	"	"	10.60	.00896	.0616	20.5	73.6	150	405	"	"	"	"	"	"
"	"	60	"	"	10.25	.00880	.0628	21.2	76.1	152	422	182	"	"	"	"	"
"	"	65	"	"	10.25	.00880	.0628	20.7	74.4	150	430	181	"	"	"	"	"
"	"	65	6.25	.000544	10.40	.00887	.0613	20.8	74.7	"	425	180	80	"	"	"	"
"	"	60	6.25	.000544	10.40	.00887	.0613	21.2	76.1	"	405	180	80	"	"	"	"
2400	1200	60	6.5	.000578	10.25	.00880	.0657	22.0	79.0	150	430	182	82	151	28.0	32.0	30.2
"	"	50	"	"	10.40	.00887	.0653	22.55	81.0	"	420	180	82	149	"	"	"
"	"	40	"	"	10.55	.00894	.0649	20.9	75.1	"	408	179	81	148	"	"	"
"	"	30	"	"	10.60	.00896	.0647	17.7	63.5	"	400	180	81	148	"	"	"
2400	1200	30	7.25	.000704	10.55	.00893	.0790	20.6	74.0	150	392	180	82	150	28.0	32.0	30.2
"	"	40	"	"	10.50	.00889	.0792	22.6	81.1	149	415	181	"	151	"	"	"
"	"	50	"	"	10.45	.00888	.0793	22.2	79.7	150	418	180	"	150	"	"	"
"	"	55	"	"	10.40	.00885	.0796	21.7	77.9	151	428	180	"	150	"	"	"
"	"	45	"	"	10.40	.00891	.0791	22.6	81.1	150	415	179	79	149	"	"	"
"	"	45	"	"	10.45	.00889	.0792	22.8	81.9	"	415	180	78	151	"	"	30.0
"	"	50	7.2	.000697	10.30	.00882	.0790	22.2	79.7	"	421	181	78	151	"	"	"
"	"	40	7.2	.000697	10.40	.00886	.0787	22.5	80.8	"	405	180	79	150	"	"	"
"	"	30	7.25	.000704	10.60	.00896	.0786	20.8	74.7	"	394	180	82	150	"	"	"
2400	1200	30	7.4	.000730	10.70	.00899	.0813	21.1	75.8	150	389	178	81	149	28.0	32.0	30.0
"	"	30	7.7	.000782	10.70	.00899	.0870	20.8	74.7	149	381	179	81	150	"	"	"
"	"	40	"	.000882	10.65	.00897	.0872	22.4	80.4	149	390	179	80	149	"	"	"
"	"	50	"	.000782	10.40	.00885	.0883	22.1	79.4	150	400	180	80	150	"	"	"
"	"	50	7.65	.000771	10.40	.00885	.0871	22.1	79.4	150	404	180	81	150			
14 February 1950 (orifice diameter 0.413")																	
2400	1200	45#	6.85	.000637	10.05	.00880	.0725	21.8	78.2	150	410	180	82	150	28.0	32.0	30.6
• "	"	"	"	"	"	"	"	21.6	77.5	"	"	"	"	"	"	"	"
• "	"	"	"	"	"	"	"	21.6	77.5	"	"	"	"	"	"	"	"
3000	1500	50#	7.8	.000801	15.95	.01099	.0729	20.5	73.6	150	435	180	85	149	28.0	32.0	30.6
• "	"	"	"	"	"	"	"	20.3	72.9	"	"	"	"	150	"	"	"
1800	900	40#	5.8	.000464	5.45	.00650	.0716	21.7	77.9	150	380	181	85	150	28.0	32.0	30.6
• "	"	"	"	"	"	"	"	21.5	77.2	"	"	180	"	"	"	"	"
960	480	35#	4.2	.000240	1.45	.00341	.0706	20.8	74.7	150	318	180	85	150	28.0	32.0	30.6
* "	"	"	"	"	"	"	.0706	20.6	73.9	"	"	"	"	"	"	"	"

• Indicator Cards Taken - Note That When Indicator Operating, Dynamometer Scale Reading
Best Power Spark Advance Was Reduced by 0.2 in Hg.

TABLE IV

2 1/2" M. I. T. G. S. Engine

Motoring Friction Data

10 January 1950 (RPM = 2400; s = 1200; p_i = 28.0; p_e = 32.0)

Time-Sec	15	30	45	60	75	165	180	195	210	
Dyn. Scale	7.8	7.6	7.9	7.8	7.7	8.0	8.0	7.9	*8.0	*FMEP = 28.7 psi
T_{wj}	176	176	176	170	170	162	161	160	160	

7 February 1950 (RPM = 2400; s = 1200; p_i = 28.0; p_e = 32.0)

Time Sec	30	60	90	120	180	240	300	360	420	480	540	600	
Dyn Scale	7.85	7.9	7.9	7.85	7.8	7.9	7.85	7.85	7.85	*7.85	7.8	7.9	*FMEP = 28.2 psi
T_{wj}	180	180	181	181	180	180	180	180	180	180	180	180	

12 February 1950 (T_i = 150; T_{wj} = 180; T_{oil} = 150; p_i = 28.0; p_e = 32.0)

RPM = 960; s = 480

Time Sec	15	30	45	60	75	90	
Dyn Scale	5.8	5.7	5.8	5.9	6.0	*5.8	*FMEP = 20.8 psi

RPM = 1200; s = 600

Time Sec	15	30	45	60	75	90	105	120	150	180	210	240	
Dyn Scale	6.0	6.1	6.4	6.0	6.5	6.0	6.5	6.0	5.9	6.4	6.4	*6.4	*FMEP = 22.95 psi

RPM = 1800; s = 900

Time Sec	15	30	45	60	75	90	105	120	150	
Dyn Scale	7.4	7.0	7.1	7.2	7.1	7.0	7.3	7.1	*7.2	*FMEP = 25.85 psi

RPM = 2400; s = 1200

Time Sec	15	30	45	60	75	90	105	120	150	
Dyn Scale	8.5	8.1	8.2	8.1	8.2	8.2	8.0	8.2	*8.1	*FMEP = 29.1 psi

RPM = 3000; s = 1500

Time Sec	15	30	45	60	75	90	105	
Dyn Scale	9.65	9.35	9.4	9.45	9.5	9.5	*9.45	*FMEP = 33.9 psi

RPM = 3400; s = 1700

Time Sec	15	30	45	60	75	90	105	120	150	165	180	195	210	
Dyn Scale	10.1	10.2	10.2	10.3	10.2	10.25	10.25	10.1	10.2	10.15	10.15	10.2	*10.2	*FMEP = 36.6 psi

TABLE V

2 1/2" M. I. T. G. S. Engine

Detonation Data

Fuel "White Gas"

21 March 1950 (orifice diameter 0.413")

RPM	s	Spark Advance OBTC	Rotometer	$\dot{m}_f$	Δp orifice	$\dot{m}_a$	F	Dyn. Scale	BMEP	T_i	T_{cyl}	T_{wj}	T_a	T_{oil}	P_i	P_e	P_a
#2400	1200	65	10.0	.001235	9.85	.0102	.1210	24.35	87.4	180	---	180	80	150	32.0	32.0	42.0
# "	"	58	10.2	.001274	10.55	.01054	.1209	25.95	93.1	"	367	"	81	"	33.0	"	42.0
# "	""	60	10.4	.001316	11.40	.01095	.1200	26.95	96.8	"	362	"	81	"	34.0	"	41.8
# "	"	65	10.65	.001371	12.25	.01135	.1209	28.35	101.8	"	356	"	80	"	35.0	"	41.85
"	"	74	11.1	.001462	14.00	.01208	.1211	30.25	108.6	"	403	"	79	"	37.0	"	41.25
"	"	57	11.75	.001605	16.80	.01330	.1209	34.75	124.9	"	388	"	80	"	40.0	"	41.95
"	"	63	11.45	.00.538	15.25	.01272	.1209	33.15	118.0	"	384	"	80	"	38.5	"	42.15
2400	1200	66	10.45	.001330	14.05	.01210	.1100	30.95	111.1	180	413	180	80	150	37.0	32.0	41.25
"	"	52	10.8	.001402	15.75	.01281	.1094	34.05	122.17	"	415	"	"	"	39.0	"	41.45
"	"	77	10.0	.001235	12.10	.01130	.1094	26.75	96.0	"	418	"	"	"	35.0	"	41.90
"	"	49	11.0	.001442	16.5	.01321	.1094	35.35	126.9	"	408	"	"	"	40.0	"	42.10
2400	1200	52	10.15	.001265	15.0	.01260	.1003	34.15	122.7	180	416	180	80	150	38.5	32.0	42.20
"	"	45	10.4	.001316	16.5	.01392	.0994	35.75	128.2	"	418	"	"	"	40.0	"	42.00
"	"	59	9.75	.001182	13.1	.01189	.0994	30.75	110.3	"	425	"	"	"	37.0	"	42.75
"	"	72	9.4	.001110	11.5	.01117	.0994	27.05	97.1	"	443	"	"	"	35.0	"	42.90
2400	1200	62	8.8	.00099	11.5	.01117	.0867	28.15	101.0	180	452	180	80.5	150	35.0	32.0	42.80
"	"	50	9.25	.001081	13.5	.01209	.0895	32.85	118.0	"	443	"	80.5	"	37.0	"	42.50
"	"	43	9.55	.001142	15.25	.01278	.0895	35.85	128.8	"	448	"	80	"	39.0	"	42.40
"	"	40	9.725	.001180	16.35	.01317	.0897	37.15	133.3	"	438	"	80	"	40.0	"	42.10
2400	1200	34.5	9.1	.001049	16.0	.01308	.0802	37.25	133.8	180	446	180	80	150	40.0	32.0	42.40
"	"	40	8.8	.000991	14.3	.01240	.0798	35.05	125.9	"	454	"	81	"	38.0	"	42.70
"	"	49	8.5	.000930	12.3	.01152	.0805	31.85	114.3	"	453	"	"	"	36.0	"	42.80
"	"	60	8.05	.000845	10.4	.01054	.0803	27.85	100.0	"	464	"	"	"	34.0	"	43.20
2400	1200	48	7.55	.000755	10.85	.01082	.0698	30.05	107.9	180	452	180	80.5	150	34.0	32.0	42.90
"	"	42	7.85	.000810	12.4	.01155	.0701	32.45	116.5	"	453	"	80	"	36.0	"	42.60
"	"	35	8.2	.000872	14.4	.01240	.0701	35.15	126.1	"	443	"	80.5	"	38.0	"	42.40
"	"	31	8.45	.000921	16.3	.01313	.0702	36.25	130.1	"	453	"	80.5	"	40.0	"	42.10
"	"	55	7.3	.000714	9.65	.01023	.0698	27.75	99.16	"	453	"	79.5	"	33.0	"	43.20
2400	1200	64	7.0	.000662	10.25	.01052	.0629	26.15	93.9	180	433	180	79	150	33.0	32.0	43.00
"	"	57.5	7.15	.000687	11.05	.01093	.0629	27.95	100.3	"	434	"	"	"	34.0	"	42.80
"	"	46	7.4	.000730	12.5	.01161	.0629	30.25	108.6	"	441	"	"	"	36.0	"	42.50
"	"	41	7.7	.000783	14.75	.01257	.0624	32.05	115.1	"	441	"	"	"	38.0	"	42.30
"	"	37.5	8.05	.000845	16.8	.01340	.0631	33.95	121.9	"	443	"	"	"	40.1	"	42.10

Incipient Detonation Occurred For All Points Except Those
Marked # For Which No Detonation Occurred.
LI Indicator Used

TABLE VI
2 1/2" M. I. T. G. S. Engine
Detonation Data
Fuel "White Gas"

29 March 1950 (orifice diameter 0.375")

RPM	s	SPARK ADVANCE °BTC	Roto-meter	$\dot{m}_f$	Δp orifice	$\dot{m}_a$	F	Dyn. Scale	BMEP	T_i	T_{cyl}	T_{wj}	T_a	T_{oil}	P_i	P_e	P_a
1000	500	43	4.45	.000269	1.9	.00384	.0701	22.85	82.0	180	352	180	85	151	30.2	32.0	43.7
"	"	32	4.65	.000296	2.25	.00418	.0707	26.25	94.2	"	351	"	"	149	32.1	"	43.7
"	"	25	4.85	.000325	2.80	.00464	.0710	29.35	105.3	"	352.5	"	"	151	34.8	"	43.4
"	"	19	5.05	.0003525	3.30	.00502	.0701	31.75	114.0	"	358	181	"	151	36.7	"	43.2
1000	500	21	5.35	.000395	3.30	.00502	.0786	32.65	117.1	180	359	180	85	149	36.7	32.0	43.2
"	"	28	5.10	.000351	2.70	.00455	.0785	29.15	104.8	"	342	179	"	151	33.9	"	43.35
"	"	41	4.75	.000310	2.05	.00397	.0782	23.65	84.9	"	352	180	"	149	30.95	"	43.5
"	"	33	4.95	.000336	2.40	.004285	.0784	26.95	96.8	"	347.5	181	86	151	32.7	"	43.5
1000	500	35	5.20	.000375	2.37	.004265	.0880	26.45	95.0	180	344	181	86	150	32.6	32.0	43.4
"	"	29	5.45	.000409	2.85	.00466	.0878	29.35	105.3	180	342	180	"	152	34.8	"	43.25
"	"	23	5.70	.000448	3.48	.00514	.0872	32.45	116.6	179	342	179	"	151	37.25	"	43.1
"	"	47	4.95	.000338	1.95	.00387	.0874	22.15	79.5	180	336	179	"	150	30.25	"	43.6
1000	500	51	5.25	.000380	1.95	.00387	.0980	22.10	79.4	180	327.5	181	86	150	30.4	32.0	43.6
"	"	43	5.50	.000417	2.40	.004285	.0975	25.25	90.6	"	331	179	86	151	32.5	"	43.25
"	"	33	5.75	.000455	2.90	.00465	.0975	28.85	103.6	"	329	180	84	150	34.6	"	43.0
"	"	28	6.05	.000505	3.59	.00520	.0975	32.45	116.6	"	325	180	84	150	37.4	"	42.9
1000	500	34	6.35	.000555	3.59	.00520	.1069	31.35	112.6	180	317.5	180	84.5	150	37.4	32.0	42.9
"	"	41	6.15	.000520	3.05	.004835	.1076	29.45	105.9	"	302	"	85	"	34.85	"	43.3
"	"	46	5.85	.000473	2.55	.004415	.1071	27.25	97.8	"	310	"	85	"	33.25	"	43.3
1000	500	45	4.15	.000235	1.95	.00387	.0614	23.45	84.2	180	338	180	84.5	150	30.1	32.0	43.4
"	"	33	4.45	.000272	2.50	.00439	.0620	27.75	99.6	"	342	"	"	"	32.7	"	43.4
"	"	24	4.65	.0002975	3.10	.00484	.0615	29.75	106.9	"	349	"	"	"	35.2	"	43.1

TABLE VII
2 1/2" M. I. T. G. S. Engine
Determination of Best Power Spark Advance Operating Supercharged
Fuel 100 Octane

4 April 1950 (orifice 0.375")

RPM	s	Spark Advance BTC	Roto-meter	$\dot{M}_f$	Δp orifice	$\dot{M}_a$	F	Dyn. Scale	BMEP	T_1	T_{cyl}	T_{wj}	T_a	T_{oil}	P_1	P_e	P_a
2400	1200	23	7.55	.000755	15.1	.0104	.0726	23.15	83.1	180	418	180	83	150	33.0	32.0	42.7
"	"	27	"	"	"	"	"	25.55	91.7	"	425	"	84	148	"	"	"
"	"	33	"	"	"	"	"	27.05	97.0	"	432	"	85	151	"	"	"
"	"	39	"	"	"	"	"	28.45	102.1	"	435	"	86	150	"	"	"
"	"	45	"	"	"	"	"	28.85	103.6	"	445	"	"	"	"	"	"
"	"	51	"	"	"	"	"	28.70	103.0	"	453	"	"	"	"	"	"
"	"	56	"	"	"	"	"	28.15	101.1	"	459	"	"	"	"	"	"
2400	1200	56	7.95	.000825	18.4	.01141	.0723	31.95	114.8	180	475	180	86.5	150	35.9	32.0	42.3
"	"	51	8.00	.000835	"	"	.0731	32.30	116.0	"	469	"	87	"	"	"	"
"	"	45	"	"	"	"	.0731	32.55	116.9	"	459	"	"	"	"	"	"
"	"	40	"	"	18.5	.01143	.0730	32.35	116.1	"	450	"	"	"	"	"	"
"	"	35	"	"	18.55	.01145	.0729	31.55	113.2	"	444	"	"	"	"	"	"
"	"	30	"	"	18.55	.01145	.0729	30.05	107.9	"	440	"	"	"	"	"	"
2400	1200	30	8.4	.000911	21.3	.01222	.0744	33.25	119.2	180	442	180	87	150	38.9	32.0	42.0
"	"	40	"	"	"	"	"	35.45	127.2	"	458	"	87	"	"	"	"
"	"	50	"	"	"	"	"	35.15	126.1	"	477	"	88	"	"	"	"
1000	500	50	4.3	.000251	1.72	.003595	.0698	20.85	74.8	181	346	180	88	150	29.5	32.0	44.9
"	"	45	"	"	"	"	"	21.85	78.4	180	343	"	"	"	"	"	"
"	"	39	"	"	"	"	"	22.80	81.8	"	336	"	"	"	"	"	"
"	"	32	"	"	"	"	"	22.95	82.4	"	331	"	"	"	"	"	44.8
"	"	25	"	"	"	"	"	22.45	80.6	"	325	"	"	"	"	"	"
"	"	20	"	"	"	"	"	21.55	77.4	"	322	"	"	"	"	"	"
1000	500	20	4.8	.000316	2.6	.00449	.0705	26.75	96.0	180	336	180	87	150	33.3	32.0	44.4
"	"	25	"	"	"	"	"	27.85	100.0	"	339	"	"	"	"	"	"
"	"	31	"	"	"	"	"	28.15	101.1	"	345	"	"	"	"	"	"
"	"	37	"	"	"	"	"	27.75	99.6	"	351	"	"	"	"	"	"
"	"	42	"	"	"	"	"	27.15	97.5	"	355	"	"	"	"	"	"
1000	500	42	5.1	.000360	3.3	.00506	.0712	30.45	109.2	180	362	180	87	150	36.2	32.0	44.1
"	"	35	"	"	"	"	"	31.65	113.8	"	356	"	"	"	"	"	"
"	"	27	"	"	"	"	"	31.75	114.0	"	349	"	"	"	"	"	"
"	"	20	"	"	"	"	"	30.75	110.4	"	342	"	"	"	"	"	44.0

TABLE VIII

2 1/2" M. I. T. G. S. ENGINE

Fuel 100 Octane

7 February 1950 (orifice diameter 0.413")

RPM	s	Spark Advance	Roto-meter	$\dot{M}_f$	ΔP orifice	$\dot{M}_a$	F	Dyn. Scale	BMEP	T_i	T_{cyl}	T_{wj}	T_a	T_{oil}	P_i	P_e	P_a
2400	1200	45	6.75	.000620	9.61	.00857	.0725	21.7	77.9	175	413	180	80	149	28.0	32.0	30.2
"	"	"	6.80	.000628	9.85	.00869	.0725	22.05	79.1	167	410	"	"	150	"	"	"
"	"	"	6.85	.000637	10.08	.00878	.0727	22.4	80.4	159	418	182	"	152	"	"	"
"	"	"	6.9	.000644	10.15	.00879	.0733	22.6	81.2	154	413	181	"	149	"	"	"
"	"	"	6.9	"	10.35	.00885	.0728	22.85	82.0	148	417	180	"	152	"	"	"
"	"	"	6.9	"	10.46	.00890	.0725	23.0	82.6	143	417	181	81	149	"	"	"
"	"	"	6.95	.000654	10.75	.00904	.0726	23.3	83.6	137	411	179	81	150	"	"	"
Effect of Variation of T_{wj} on BMEP																	
2400	1200	45	6.85	.000637	10.25	.00880	.0723	22.7	81.5	150	415	181	81	150	28.0	32.0	30.2
"	"	"	6.9	.000645	10.35	.00885	.0729	22.7	81.5	151	410	172	"	"	"	"	"
"	"	"	6.9	.000645	10.40	.00889	.0726	22.8	81.8	150	403	162	"	"	"	"	"
"	"	"	6.85	.000637	10.50	.00892	.0715	22.9	82.2	"	393	150	"	151	"	"	"
"	"	"	"	"	10.15	.00876	.0727	22.65	81.4	"	430	198	"	"	"	"	"
"	"	"	"	"	10.15	.00876	.0727	22.85	82.0	"	427	192	"	"	"	"	"

TABLE IX

4" M. I. T. C. S. Engine

Fuel 100 Octane

RPM	s	Spark Advance	Roto-meter	$\dot{M}_f$	Δp orifice	$\dot{M}_a$	F	Dyn. Scale	BMEP	T_i	T_{cyl}	T_{wj}	T_a	T_{oil}	P_i	P_e	P_a
6 December 1949 (orifice diameter 0.614")																	
1200	960	30	8.75	.00132	1.75	.0181	.0730	27.25	89.5	151	393	180	74	151	28.0	32.0	29.8
"	"	46	8.70	.001315	"	"	"	26.55	87.0	151	395	180	"	151	"	"	"
"	"	26	8.70	.001315	"	"	"	27.0	88.5	150	392	181	"	150	"	"	"
"	"	20.5	8.65	.00131	"	"	.0725	26.15	85.8	151	386	181	"	"	"	"	"
"	"	35.	8.75	.00132	1.73	.0180	.0730	27.35	89.6	150	396	180	"	"	"	"	"
"	"	41	8.70	.001315	1.73	.0180	.0730	27.15	89.0	150	403	180	"	"	"	"	"
13 December 1949 (orifice diameter 0.614")																	
1500	1200	40	10.4	.00165	14.45	.0228	.0723	27.4	90.0	152	426	182	84	150	28.0	32.0	30.0
"	"	30	10.4	.00165	14.55	.02285	.0722	27.1	89.0	152	404	180	"	"	"	"	"
"	"	20	10.3	.00163	14.70	.02295	.0711	25.1	82.4	151	405	180	"	"	"	"	"
"	"	20	10.5	.00166	14.70	.02295	.0724	25.05	82.15	"	402	179	"	"	"	"	"
"	"	45	10.6	.00170	14.45	.0228	.0745	26.75	87.7	"	428	180	"	"	"	"	"
"	"	45	10.5	.00166	14.40	.02275	.0730	26.9	88.4	150	432	181	"	"	"	"	"
1700	1360	45	11.6	.00191	18.85	.0261	.0731	27.1	89.0	149	449	182	84	150	28.0	32.0	30.0
"	"	40	"	"	18.85	"	"	27.3	89.5	150	439	"	"	150	"	"	"
"	"	35	"	"	18.95	"	"	27.5	90.3	"	434	"	"	151	"	"	"
"	"	30	"	"	19.10	.0262	.0729	27.05	88.8	"	429	"	"	151	"	"	"
"	"	20	"	"	19.25	.0263	.0727	24.45	80.3	"	418	"	"	150	"	"	"
9 February 1959 (orifice diameter 0.614")																	
1500	1200	31	10.2	.00161	14.5	.0228	.0705	26.9	88.2	150	415	180	80	149	28.0	32.0	30.0
"	"	35	10.6	.00169	14.6	.0229	.0737	27.3	89.5	149	410	"	"	150	"	"	"
"	"	41	10.5	.00167	14.5	.0228	.0731	27.2	89.2	150	420	"	"	"	"	"	"
"	"	45	10.5	.00169	14.45	.0228	.0731	27.1	88.8	"	430	"	"	"	"	"	"
"	"	49	10.3	.00163	14.1	.0225	.0725	26.5	86.9	"	435	182	"	"	"	"	"
"	"	53	10.3	.00163	14.1	.0225	.0725	26.1	85.6	"	440	183	"	"	"	"	"
"	"	55	10.5	.00167	14.25	.0227	.0735	25.6	83.9	151	450	182	"	"	"	"	29.9
"	"	61	"	"	14.1	.0225	.0742	24.8	81.3	150	455	180	"	"	"	"	"
"	"	45	"	"	14.5	.0228	.0731	27.2	89.2	"	430	"	"	"	"	"	"
"	"	37	10.45	.00166	14.4	.0227	.0731	27.5	90.2	"	415	"	"	"	"	"	"
1200	960	37	8.7	.00131	9.0	.0180	.0731	27.4	89.8	149	395	180	80	150	28.0	32.0	29.9
"	"	30	"	"	8.9	.0179	.0732	27.3	89.5	150	390	"	"	"	"	"	"
"	"	43	"	"	8.9	.0179	.0732	26.6	87.3	150	400	"	"	"	"	"	"
600	480	43	4.8	.000645	2.2	.00890	.0725	24.3	79.6	150	345	180	80	150	28.0	32.0	29.9
"	"	42.5	"	"	"	"	"	25.6	83.9	"	335	"	"	"	"	"	"
"	"	26	"	"	"	"	"	26.5	86.9	"	330	"	"	"	"	"	"
"	"	20	"	"	"	"	"	25.8	84.6	"	320	"	"	"	"	"	"
"	"	27	"	"	"	"	"	26.2	85.9	"	325	"	"	"	"	"	"
16 February 1950 (orifice diameter 0.614")																	
1500	1200	40	10.5	.00167	14.25	.0228	.0732	27.5	90.2	150	423	181	80	150	28.0	32.0	29.9
1500	1200	40	10.2	.00165	14.2	.02265	.0728	27.5	90.2	"	428	180	79	150	28.0	32.0	29.9

TABLE X
4" E. I. T. O. S. Engine
Fuel 100 Octane

7 April 1950 (orifice diameter 0.614")

RPM	s	Spark Advance	Roto-meter	$\dot{M}_f$	Δp orifice	$\dot{M}_a$	F	Dyn. Scale	BMEP	T_i	T_{cyl}	T_{wj}	T_a	T_{oil}	P_i	P_e	P_a
1500	1200	31	10.6	.00169	14.65	.02315	.0730	27.85	91.2	150	394	180	80	150	28.0	32.0	30.15
"	"	24	"	"	14.60	.02315	.0730	26.70	87.6	150	390	180	"	150	"	"	"
"	"	40	"	"	14.40	.02300	.0734	27.95	91.7	151	411	181	"	149	"	"	"
"	"	45	10.5	.00167	14.30	.02290	.0730	27.90	91.5	150	414	180	"	150	"	"	"
"	"	52	"	"	"	"	"	27.20	89.1	149	425	"	"	149	"	"	"
* "	"	39	"	"	"	"	"	28.0	91.8	150	405	"	"	150	"	"	"
1800	1440	51	12.05	.00203	21.1	.0278	.0731	26.95	88.5	149	465	180	80	150	28.0	32.0	30.15
"	"	45	"	"	21.1	.0278	.0731	27.90	91.5	149	453	"	"	"	"	"	"
"	"	37	"	"	21.0	.02775	.0732	27.95	91.7	150	440	"	"	"	"	"	"
"	"	30	"	"	21.1	.0278	.0731	27.30	89.6	"	430	"	"	"	"	"	"
* "	"	41	"	"	21.1	.0278	.0731	28.15	92.5	"	447	"	"	"	"	"	"
900	720	41	6.9	.00097	4.8	.01327	.0732	27.25	89.3	150	352	179	80	151	28.0	32.0	30.15
"	"	49	"	"	4.75	.01320	.0734	26.2	85.9	"	366	180	"	150	"	"	"
"	"	29	"	"	4.8	.01327	.0732	27.6	90.5	"	350	"	"	"	"	"	"
"	"	35	"	"	"	"	"	27.8	91.1	"	351	"	"	"	"	"	"
* "	"	32	"	"	"	"	"	27.8	91.1	"	353	"	"	"	"	"	"
1000	800	32	8.5	.00127	5.5	.01745	.0728	33.4	109.6	180	384	180	80	150	32.6	32.0	45.15
"	"	25	"	"	5.5	.01745	.0728	32.85	107.6	"	375	"	"	"	"	"	"
"	"	40	"	"	5.45	.01740	.0731	33.0	108.3	"	392	"	"	"	"	"	"
"	"	36	"	"	"	"	"	33.25	109.0	"	387	"	"	"	"	"	"
"	"	29	"	"	"	"	"	33.2	108.8	"	380	"	"	"	"	"	"
1500	1200	29	11.7	.00193	14.6	.02645	.0730	33.3	109.6	180	425	180	80	150	32.8	32.0	39.15
"	"	36	"	"	"	"	"	34.0	111.6	"	432	"	"	"	"	"	"
"	"	43	"	"	"	"	"	33.75	110.7	"	444	"	"	"	"	"	"
"	"	48	"	"	14.4	.0263	.0734	33.1	108.6	181	455	"	"	"	"	"	"

*Indicator Cards Taken

Motoring Friction 16 February 1950 P_e = Pi = atmospheric

RPM	s	Time ‡	Dyn. Scale	MEP	$T_{wJ(av)}$	$T_{oil(av)}$	Remarks
1500	1200	2 00	-7.6	24.95	182	153	After firing
2000	1600	--	-8.5	27.85	182	151	Before Firing
1800	1440	2 00	-8.35	27.4	180	150	After Firing
1200	960	--	-6.35	20.8	181	150	Before Firing
1200	960	2 00	-6.50	21.3	180	150	After Firing
1050	840		-6.60	21.65	185	150	After Firing

‡ Time required after ignition
is cut to stabilize

TABLE XI

4" M. I. T. G. 2. ENGINE

Detonation Data

Fuel-White Gas

27 March 1950 (orifice diameter 0.614")

RPM	s	Spark Advance	Roto-meter	$\dot{M}_f$	Δp orifice	M_a	F	Dyn. Scale	BMEP	T_i	T_{cyl}	T_{wj}	T_a	T_{oil}	P_i	P_e	P_a
1500	1200	25	9.3	.001418	10.85	.0236	.0600	26.2	86.0	180	410	181	75	150	29.7	32.0	41.85
"	"	21	10.0	.001570	14.1	.0262	.0599	28.4	93.2	181	417	182	"	150	32.45	"	39.7
"	"	18	10.65	.00170	17.85	.0287	.0593	29.5	96.8	"	417	181	"	151	34.9	"	37.6
"	"	15	11.3	.00183	19.5	.0304	.0602	32.1	105.4	"	430	181	"	150	36.5	"	38.5
"	"	42.5	8.5	.00127	7.85	.0212	.05985	24.15	79.2	180	420	180	"	150	27.15	"	46.75
1500	1200	32	11.6	.00191	12.4	.0240	.0796	29.1	95.5	180	416	182	75	151	30.2	31.9	37.8
"	"	26	12.4	.002085	13.0	.0260	.0802	32.35	106.1	"	422	181	"	150	32.35	32.0	42.4
"	"	18	13.15	.00226	16.15	.02825	.0800	33.7	110.6	"	418	180	"	151	34.6	"	40.3
"	"	15	13.70	.00239	18.85	.02985	.0801	34.9	114.6	"	423	181	"	150	36.4	"	38.5
"	"	42	11.15	.00185	9.075	.02265	.0802	27.3	89.4	"	410	480	"	"	28.6	32.0	46.05
1500	1200	19	14.8	.002665	19.0	.0300	.0890	35.5	116.6	180	417	180	75	151	36.4	32.0	38.5
"	"	22.5	14.3	.00254	16.15	.0283	.0897	34.2	112.2	181	410	180	"	150	34.75	32.0	40.35
"	"	30.5	13.3	.00229	12.8	.0258	.0889	32.1	105.4	180	414	181	"	151	32.1	31.9	42.5
"	"	41	12.7	.00215	10.8	.0241	.0891	29.3	96.2	180	419	181	"	151	30.5	32.0	43.9
1500	1200	61	13.8	.00241	10.9	.02425	.0994	26.35	86.4	180	423	180	75	151	30.5	32.0	43.9
"	"	38.5	14.6	.00261	13.25	.0261	.1000	31.9	104.7	179	408	180	"	150	32.4	"	42.2
"	"	28	15.4	.00281	15.95	.02805	.1001	34.5	113.3	181	400	182	"	"	34.4	"	40.2
"	"	23.5	16.2	.00301	19.05	.02995	.0995	35.65	117.0	181	407	181	"	"	36.35	"	38.4
29 March 1950 (orifice diameter 0.614")																	
1500	1200	32.5	18.25	.00355	18.55	.0296	.120	34.1	111.8	179	360	180	86	150	35.3	32.0	38.55
"	"	41.0	17.7	.00340	16.5	.0282	.1205	32.8	107.6	180	355	"	"	"	33.85	"	39.35
"	"	46.5	17.2	.00329	15.4	.0273	"	32.5	106.7	179	355	"	"	"	32.85	"	39.60
"	"	31.0	19.15	.00379	20.5	.0315	"	36.2	118.7	180	360	"	"	"	37.3	"	39.60

Incipient Detonation as Determined by Li Indicator Occurred

at All Above Points

TABLE XII
4" M. I. T. G. S. Engine
Detonation Data
Fuel - White Gas

11 April 1950 (orifice diameter 0.614")

RPM	s	Spark Advance	Roto-meter	$\dot{M}_f$	ΔP orifice	$\dot{M}_a$	F	Dyn. Scale	BMEP	T_i	T_{cyl}	T_{wj}	T_a	T_{oil}	P_i	P_e	P_a
1000	800	18	8.7	.00131	5.9	.01791	.0731	32.15	105.5	180	375	180	84	150	32.25	32.0	45.10
"	"	24	8.1	.00111	4.8	.01635	.0728	30.15	98.8	"	378	"	"	"	30.65	"	46.10
"	"	29	7.4	.00105	3.65	.01441	.0728	27.85	91.2	"	376	"	"	"	29.1	"	47.30
"	"	36	6.8	.000953	2.95	.01305	.0730	25.55	83.7	"	380	"	"	"	27.6	"	48.0
1000	800	38	6.0	.00082	3.2	.01356	.0605	23.65	77.5	180	362	180	84	150	28.0	32.0	47.75
"	"	24	7.3	.00103	5.25	.0170	.0606	30.10	98.6	180	382	"	"	"	31.35	"	45.55
"	"	20	7.8	.00113	6.45	.01862	.0607	32.0	104.7	179	384	"	"	"	33.3	"	44.5
"	"	37.5	6.1	.00084	3.4	.01394	.0602	24.7	81.0	180	367	"	"	"	28.5	"	47.45
1000	800	43	8.1	.001185	3.4	.01394	.0850	25.8	84.6	180	379	180	84	151	28.6	32.0	47.45
"	"	29	9.4	.00144	5.2	.0169	.0852	30.5	100.0	179	376	180	"	151	31.5	"	45.6
"	"	23	10.15	.001595	6.45	.01875	.0851	33.3	109.2	180	377	181	"	150	33.4	"	44.5
1000	800	30	11.4	.00187	6.45	.01875	.0997	32.7	107.4	181	361	180	85	150	33.45	32.0	44.5
"	"	23.5	12.25	.00205	8.1	.0205	.1000	35.75	117.4	180	360	"	"	150	35.85	"	43.15
"	"	45	10.0	.001568	4.4	.01569	.1001	27.4	89.8	180	358	"	"	151	30.15	"	46.4
1500	1200	52.5	9.3	.00142	7.8	.0203	.0700	24.1	79.1	180	442	180	85	151	27.3	32.0	43.7
"	"	45.5	9.45	.001448	8.25	.0207	.0699	25.75	84.4	"	432	179	"	150	27.75	"	43.1
"	"	34.0	10.3	.001628	10.65	.02325	.0699	28.9	94.8	"	436	180	"	"	29.70	"	41.5
"	"	27	10.8	.00174	12.7	.02485	.0700	30.8	100.9	"	433	"	"	"	30.8	"	40.3
"	"	16	12.0	.00200	17.8	.02855	.0700	32.9	107.9	"	430	"	"	"	33.95	"	38.0

Incipient Detonation as Determined by Li Indicator Occurred
at all above points.

TABLE XIII

6" M. I. T. C. S. Engine
Fuel - 100 Octane

RPM	a	Spark Advance °BTC	Roto-meter	$\dot{M}_f$	Δp orifice	$\dot{M}_a$	F	Dyn. Scale	BMEP	T_1	T_{cyl}	T_{wj}	T_{air}	T_{oil}	P_1	P_e	P_a
23 November 1949 (orifice diameter 0.920")																	
800	960	30	4.8	.002625	8.90	.04010	.0653	21.2	82.50	152	462	180	85	152	28.0	32.0	29.85
"	"	30	4.4	.002450	9.10	.04030	.0607	21.1	82.10	150	435	179	90	151	"	"	"
"	"	35	5.1	.002740	8.75	.03940	.0696	23.3	90.60	149	487	180	92	150	"	"	"
"	"	35	5.4	.002875	8.70	.03920	.0734	23.45	91.20	150	490	180	93	152	"	"	"
"	"	35	5.8	.003050	8.75	.03930	.0777	23.6	91.75	150	485	180	93	152	"	"	"
"	"	35	6.25	.003240	8.85	.03955	.0819	23.85	92.75	150	477	180	94	152	"	"	"
"	"	30	6.25	.003240	8.85	.03955	.0819	24.15	94.00	150	465	181	94	153	"	"	"
"	"	40	4.4	.002440	8.95	.03975	.0614	21.85	85.00	149	458	180	96	152	"	"	"
"	"	30	4.4	.002440	8.95	.03975	.0614	21.85	85.00	150	446	180	96	152	"	"	"
"	"	20	4.4	.002440	9.1	.04000	.0609	20.15	78.40	150	430	180	96	152	"	"	"
"	"	20	4.45	.002480	9.1	.04000	.0618	20.55	80.00	150	433	180	96	152	"	"	"
"	"	10	4.45	.002480	9.2	.04035	.0615	16.65	64.70	150	424	180	97	152	"	"	"
"	"	35	7.1	.003580	8.9	.03960	.0904	23.75	92.40	150	456	180	98	152	"	"	"
8 December 1949 (orifice diameter 0.920")																	
800	960	40	4.9	.002680	8.8	.04030	.0663	22.85	89.0	151	480	180	80	150	28.0	32.0	29.9
"	"	35	"	"	"	.04030	.0663	23.35	90.8	152	477	179	"	"	"	"	"
"	"	30	"	"	"	"	"	22.95	89.2	149	456	180	"	"	"	"	"
"	"	25	"	"	8.85	.04040	.0662	22.15	86.1	150	450	"	"	"	"	"	"
"	"	20	"	"	"	"	"	21.75	84.5	151	449	"	"	"	"	"	"
"	"	45	"	"	8.8	.04030	.0663	22.65	88.1	151	486	"	"	"	"	"	"
"	"	50	"	"	8.75	.04020	. "	21.95	85.4	150	495	181	"	"	"	"	"
9 December 1949 (orifice diameter 0.920")																	
800	960	50	5.5	.00292	8.45	.0400	.0730	21.75	84.5	150	524	180	80	150	28.0	32.0	30.3
"	"	45	"	"	8.50	.0401	.0728	23.15	90.0	152	505	"	"	150	"	"	"
"	"	40	"	"	8.50	.0401	.0728	23.35	90.8	151	496	"	"	153	"	"	"
"	"	35	"	"	8.60	.0403	.0725	23.85	92.8	151	487	"	"	151	"	"	"
"	"	30	"	"	8.70	.0406	.0719	24.15	93.9	150	485	"	"	150	"	"	"
"	"	25	"	"	8.80	.0408	.0716	24.25	94.4	"	475	"	"	"	"	"	"
"	"	25	5.6	.00295	8.70	.0405	.0728	24.15	93.9	"	475	"	"	"	"	"	"
"	"	20	"	"	8.85	.0409	.0721	23.75	92.4	"	466	"	"	149	"	"	"
"	"	30	"	"	8.70	.0406	.0727	24.45	95.1	"	480	"	"	149	"	"	"
"	"	35	5.5	.00292	8.70	.0406	.0719	24.45	95.1	151	495	179	"	151	"	"	"
"	"	40	"	"	8.60	.0403	.0725	23.55	91.5	150	506	178	"	152	"	"	"
"	"	50	"	"	8.50	.0401	.0728	22.20	86.4	150	516	179	"	150	"	"	"
"	"	45	"	"	8.50	.0401	.0728	23.25	90.5	151	506	178	"	150	"	"	"

TABLE XIV
6" M. I. T. G. S. Engine
Fuel 100 Octane

	RPM	s	Spark Advance BTC	Roto-meter	$\dot{M}_f$	ΔP orifice	$\dot{M}_a$	F	Dyn. Scale	BMEP	T_1	T_{cyl}	T_{wj}	T_a	T_{oil}	P_1	P_e	P_a
							13 December 1949 (orifice diameter 0.920")											
	800	960	45	6.1	.00317	8.8	.0400	.0790	22.4	87.1	150	500	180	84	151	28.0	32.0	29.9
	"	"	55	5.95	.00312	8.6	.0397	.0790	19.9	77.4	"	530	"	"	152	"	"	"
	"	"	30	6.1	.00317	8.9	.0402	.0787	23.7	92.2	"	480	"	"	150	"	"	"
	"	"	20	6.15	.00319	9.0	.0405	"	22.9	89.1	"	460	"	"	"	"	"	"
	"	"	39	6.05	.00316	8.85	.0401	"	23.25	90.5	"	495	"	"	"	"	"	"
	1000	1200	39	6.7	.00343	14.35	.0511	.0670	23.4	91.0	149	535	180	85	150	28.0	32.0	29.85
	"	"	49	6.6	.00339	14.10	.0506	.0670	22.5	87.5	150	560	"	"	"	"	"	"
	"	"	30	6.7	.00343	14.5	.0514	.0666	23.4	87.1	"	505	"	"	"	"	"	"
	"	"	18	6.7	.00343	14.6	.0515	"	21.4	83.3	"	495	"	"	"	"	"	"
	"	"	45	6.65	.00340	14.2	.0510	"	23.2	90.3	"	545	"	"	"	"	"	"
	1000	1200	45	7.4	.00372	14.1	.0506	.0734	23.1	90.0	150	550	180	84	150	28.0	32.0	29.85
	"	"	35	7.5	.00376	14.4	.0512	.0734	24.2	94.1	"	570	"	"	"	"	"	"
	"	"	25	7.5	.00376	14.5	.0514	.0732	24.2	94.1	"	510	"	"	"	"	"	"
	"	"	15	7.6	.00381	14.65	.0517	.0736	22.4	87.1	"	495	"	"	"	"	"	"
	"	"	30	7.55	.00378	14.45	.0513	.0736	24.5	95.3	"	520	"	"	"	"	"	"
	1000	1200	30	8.3	.00413	14.5	.0514	.0804	24.4	95.0	150	510	180	83	150	28.0	32.0	29.85
	"	"	20	8.35	.00415	14.6	.0515	.0806	23.65	92.1	"	490	"	"	"	"	"	"
	"	"	35	8.25	.00410	14.4	.0512	.0800	24.05	93.8	"	520	"	"	"	"	"	"
	"	"	45	8.15	.00406	14.2	.0510	.0798	22.7	88.3	"	550	"	"	"	"	"	"
							10 January 1950 (orifice diameter 0.614")											
	400	480	45	1.8	.0014	10.3	.01935	.0723	19.15	74.5	151	447	180	79	150	28.0	32.0	30.15
	"	"	33	1.8	.0014	10.5	.01952	.0718	21.75	84.5	150	426	179	80	"	"	"	"
	"	"	26	1.85	.00142	10.75	.01972	.0721	22.55	87.6	"	420	180	"	"	"	"	"
	"	"	20	"	"	10.8	.01978	.0718	22.75	88.5	"	410	"	"	"	"	"	"
	"	"	15	"	"	10.65	.01965	.0722	22.45	87.4	"	405	"	"	"	"	"	"
	"	"	25	"	"	10.7	.01968	.0722	22.65	88.1	"	415	"	"	"	"	"	"
	400	480	25	1.6	.00132	10.6	.01961	.0674	22.55	87.6	150	413	180	80	152	28.0	32.0	30.1
	"	"	14	1.6	.00132	10.7	.01968	.0671	21.55	83.6	"	395	"	"	149	"	"	"
	"	"	20	1.65	.00133	10.75	.01972	.0675	22.45	87.4	"	404	"	"	150	"	"	"
	"	"	29	1.55	.00130	10.65	.01965	.0662	21.85	85.0	"	412	"	"	150	"	"	"
	400	480	29	2.15	.00154	10.65	.01965	.0784	22.25	86.5	150	419	180	80	150	28.0	32.0	30.0
	"	"	22	"	"	10.75	.01972	.0781	22.95	89.2	149	403	179	80	148	"	"	"
	"	"	15	"	"	10.75	.01972	.0781	22.55	87.6	150	396	180	81	150	"	"	"
	"	"	26	"	"	10.7	.01968	.0779	22.55	87.6	150	415	180	82	153	"	"	"
	400	480	26	1.85	.00142	10.7	.01968	.0722	22.55	87.6	150	415	180	82	150	28.0	32.0	30.0
	"	"	22	"	"	"	"	"	22.85	89.0	"	411	"	"	"	"	"	"
	"	"	32	"	"	"	"	"	21.85	85.0	"	428	"	"	"	"	"	"
	"	"	18	"	"	10.8	.01978	.0720	22.75	88.5	"	405	"	"	153	"	"	"

TABLE XV
6" M. I. T. C. S. ENGINE
Fuel 100 Octane

7 February 1950

RPM	s	Spark Advance	Roto-meter	$\dot{M}_f$	Δp Orifice	$\dot{M}_a$	F	Dyn. Scale	BMEP	T_i	T_{cyl}	T_{wj}	T_{air}	T_{oil}	P_i	P_e	P_a
*400	480	20	1.87	.00143	10.5	.0194	.0730	23.3	90.5	150	415	180	80	150	28.0	32.0	30.2
*800	960	30	5.45	.00290	8.8	.0406	.0715	25.7	100.0	150	510	180	80	150	28.0	32.0	30.2
*1000	1200	30	7.60	.00381	14.15	.0515	.0740	25.7	100.0	150	540	180	82	150	28.0	32.0	30.2
*1200	1440	34	9.25	.00456	21.0	.0624	.0733	24.5	95.0	150	565	180	82	150	28.0	32.0	30.2

9 March 1950 (orifice diameter 1.227")

RPM	s	Spark Advance	Roto-meter	$\dot{M}_f$	Δp Orifice	$\dot{M}_a$	F	Dyn. Scale	BMEP	T_i	T_{cyl}	T_{wj}	T_{air}	T_{oil}	P_i	P_e	P_a
1500	1880	30	11.2	.00547	9.55	.0754	.0726	22.10	86.0	150	575	180	80	150	28.0	32.0	29.6
"	"	35	11.2	.00547	9.55	.0754	.0726	22.45	87.4	"	"	"	"	"	"	"	"
* "	"	40	11.0	.00538	9.40	.0746	.0721	22.95	89.2	"	"	"	"	"	"	"	"
* "	"	44	11.0	.00538	9.40	.0746	.0721	22.75	88.5	"	"	"	"	"	"	"	"

*Indicator Cards Taken
400 RPM Run-0.614" Orifice
800, 1000, and 1200 RPM Runs-0.920" Orifice

14 February 1950 (orifice diameter 0.920")

RPM	s	Spark Advance	Roto-meter	$\dot{M}_f$	Δp Orifice	$\dot{M}_a$	F	Dyn. Scale	BMEP	T_i	T_{cyl}	T_{wj}	T_{air}	T_{oil}	P_i	P_e	P_a
700	840	30	4.65	.00255	6.45	.03515	.0727	23.5	91.4	150	485	180	80	150	28.0	32.0	30.6
"	"	36	"	"	6.45	.03515	.0727	23.2	90.1	150	495	"	"	"	"	"	"
"	"	25	"	"	6.50	.03520	.0725	23.7	92.0	151	480	"	"	"	"	"	"
"	"	19.5	"	"	6.60	.03550	.0720	23.2	90.1	151	470	"	"	"	"	"	"
"	"	13	"	"	6.60	.03550	.0720	21.9	85.0	152	460	"	"	"	"	"	"
"	"	27	"	"	6.50	.03520	.0725	24.1	93.6	150	485	"	"	"	"	"	"
1000	1200	27	7.5	.00376	14.05	.0516	.0728	24.75	96.1	152	530	180	86	150	28.0	32.0	30.5
"	"	32	7.5	.00376	14.0	.0516	.0728	25.15	97.9	150	540	"	"	"	"	"	"
"	"	37	7.4	.00373	13.9	.0514	.0728	24.9	87.0	"	555	"	"	"	"	"	"
"	"	42	7.45	.00374	13.8	.0513	.0730	24.2	94.2	"	565	"	"	"	"	"	"
"	"	35	7.45	.00374	14.05	.0516	.0730	25.2	98.0	"	550	"	"	"	"	"	"
1300	1560	35	10.05	.00493	24.1	.0669	.0736	24.7	96.1	150	575	180	86	150	28.0	32.0	30.4
"	"	39	10.0	.00491	24.0	.0669	.0735	24.7	96.1	"	"	"	"	"	"	"	"
"	"	42	10.0	.00491	24.0	.0669	.0735	24.5	95.3	"	"	"	"	"	"	"	"
"	"	47	10.0	.00491	23.8	.0664	.0741	23.7	92.2	"	"	"	"	"	"	"	"
"	"	29	10.1	.00496	24.4	.0671	.0742	25.2	98.0	"	"	"	"	"	"	"	"
"	"	34	10.0	.00491	24.2	.0669	.0735	25.35	98.5	"	"	"	"	151	"	"	"
"	"	37	10.0	.00491	24.1	.0669	.0735	25.2	98.0	"	"	"	"	150	"	"	"
400	480	22	1.9	.00144	2.0	.0198	.0727	23.3	90.5	150	405	180	86	148	28.0	32.0	30.55

TABLE XVI
6" M. I. T. G. S. Engine
Motoring Friction Data

10 January 1950 (RPM = 400; s = 480; p_e = p_i = 30.2 in. Hg)

#Time-Sec.	60	90	120	150	180	210	240	270	300	330	360	390	420	450	480	510	540
Dyn. Scale	4.35	4.25	4.25	4.25	4.35	4.4	4.4	4.5	4.3	4.5	4.4	4.45	4.45	4.45	4.45	4.45	4.45*
T_{wj}	180	180	180	175	175	175	175	176	176	177	178	178	178	178	178	178	178
T_{oil}	150	150	150	152	150	152	153	153	153	152	152	151	151	151	151	151	151
T_{cyl}	235	222	212	205	200	195	193	192	190	190	190	190	190	190	190	190	190
T_i	150	150	150	150	150	150	147	147	148	149	149	149	150	150	150	150	150

*FMEP = 17.3 psi

7 February 1950

RPM = 1200; s = 1440; p_e = p_i = 30.2 in. Hg

#Time-sec	30	60	90	120	150	180	210	240	270	300
Dyn. Scale	6.0	7.0	7.5	7.5	6.8	6.7	6.8	6.8	6.8	6.8*
T_{wj}	180	180	180	180	182	182	183	184	185	185
T_{oil}	150	150	150	150	150	150	152	154	155	155
T_i	150	150	150	150	150	150	150	150	150	150

*FMEP = 26.45 psi

RPM = 800; s = 960; p_e = p_i = 30.2 in. Hg

#Time-sec	30	60	90	120	150	180	210	240	270	300
Dyn. Scale	4.6	4.7	4.7	5.0	5.1	5.1	5.1	5.1	5.1	5.1*
T_{wj}	180	180	180	180	180	180	180	180	180	180
T_{oil}	145	145	145	145	145	145	145	145	145	145
T_i	150	150	150	150	150	150	150	150	150	150

*FMEP = 19.8 psi

RPM = 400; s = 480; p_e = p_i = 30.2 in. Hg
T_i = 150; T_{wj} = 180; T_{oil} = 150

After five minutes Dyn. Scale = 4.1; FMEP = 16.0 psi

14 February 1950

RPM	s	T_{wj}	T_i	T_{oil}	Dyn. Scale	FMEP	REMARKS
400	480	180	150	150	4.4	17.1	Reading taken 5 minutes after ignition cut.
800	960	178	146	146	5.9	22.95	Reading Taken before firing.
800	960	189	155	152	5.7	22.15	Reading taken 5 minutes after ignition cut.
1200	1440	185	155	152	7.2	28.0	Reading taken before firing.
1200	1440	179	152	152	6.9	26.85	Reading taken 5 minutes after ignition cut.

#Time after ignition cut off.

TABLE XVII
6" M. I. T. G. S. Engine
DETONATION DATA
Fuel - "White Gas"

RPM	s	Spark Advance	Roto-meter	$\dot{M}_f$	ΔP orifice	$\dot{M}_a$	F	Dyn. Scale	BMEP	T_i	T_{cyl}	T_{wj}	T_{air}	T_{oil}	P_i	P_e	P_a
14 March 1950 (orifice diameter 0.920")																	
1000	1200	27	5.45	.00290	8.4	.0396	.0731	16.45	64.0	151	514	180	79	152	22.9	32.0	30.0
"	"	14	6.65	.00342	11.65	.0467	.0731	19.85	77.2	150	513	182	79	152	25.95	32.0	30.0
"	"	10	7.55	.00379	14.2	.0515	.0736	22.25	86.5	150	512	180	79	150	28.0	32.0	30.0
#1000	1200	10	3.95	.00227	5.15	.0313	.0725	9.2	35.8	151	452	180	79	150	18.8	32.0	30.0
# "	"	27	"	"	"	"	"	11.8	45.9	150	468	"	"	151	18.8	"	"
# "	"	35	"	"	"	"	"	11.85	46.1	149	480	"	"	151	18.9	"	"
# "	"	45	"	"	"	"	"	11.5	44.7	150	500	"	"	150	18.95	"	"
# "	"	40	"	"	"	"	"	11.7	45.5	150	490	"	"	150	18.95	"	"
1000	1200	32	5.00	.00270	7.30	.0639	.0730	15.1	58.7	150	505	180	79	150	21.8	32.0	30.0
"	"	16.5	6.20	.00322	10.35	.0439	.0734	18.45	71.7	"	508	"	"	"	24.8	"	"
"	"	20.5	5.70	.00300	9.05	.0411	.0730	17.25	67.0	"	505	"	"	"	23.5	"	"
16 March 1950 (orifice diameter 0.920")																	
1000	1200	14	9.65	.00476	14.9	.0526	.0905	24.25	94.4	150	480	180	80	150	28.5	32.2	30.0
"	"	19	8.45	.00420	11.7	.0466	.0901	21.25	82.6	"	475	"	"	"	25.9	32.0	"
"	"	23	8.05	.00402	10.7	.0446	.0901	20.15	78.4	"	480	"	"	"	25.1	32.1	"
"	"	28	7.55	.00382	9.65	.0423	.0902	18.75	73.0	"	480	"	"	"	24.1	32.0	"
#1000	1200	28	7.0	.00355	8.35	.0394	.0902	17.05	66.3	150	470	180	80	150	22.8	32.1	30.0
# "	"	45	6.95	.00353	8.25	.0392	.0901	15.65	60.9	151	505	"	"	"	23.0	"	"
# "	"	51	6.90	.00351	8.25	.0392	.0897	15.65	60.9	150	515	"	"	"	23.0	"	"
1000	1200	36	8.7	.00431	9.9	.0429	.1005	18.85	73.3	150	480	180	80	150	24.5	31.95	30.0
"	"	25.5	9.3	.00458	11.35	.0460	.0997	20.85	81.1	151	470	"	"	"	25.8	32.2	"
"	"	19.5	10.3	.00506	13.75	.0506	.1000	23.25	90.5	150	470	"	"	"	27.6	32.05	"
"	"	17.5	10.85	.00530	15.05	.0531	.0999	24.65	96.0	150	465	"	"	"	28.5	31.9	"
1000	1200	19.5	11.9	.00582	15.05	.0530	.1100	24.05	93.5	150	450	180	80	150	28.5	32.0	30.0
"	"	23	11.25	.00550	13.35	.0499	.1100	22.45	87.4	"	445	"	"	"	27.3	32.1	"
"	"	24	11.25	.00550	13.35	.0499	.1100	22.65	88.1	"	445	"	"	"	27.3	32.1	"
"	"	29.5	10.2	.00500	11.1	.0454	.1100	19.95	77.5	"	440	"	"	"	25.4	31.95	"
#1000	1200	29.5	9.3	.00460	9.5	.0421	.1092	17.75	69.1	150	425	180	80	150	23.9	31.95	30.0
# "	"	40	9.3	.00460	9.45	.0420	.1095	17.65	68.7	"	455	"	"	"	23.9	31.8	"
1000	1200	31	10.15	.00498	10.95	.0452	.1100	19.65	76.5	150	445	180	80	150	25.25	32.1	30.0

INCIPIENT DETONATION OCCURRED FOR ALL POINTS EXCEPT THOSE
MARKED # FOR WHICH NO DETONATION OCCURRED.

TABLE XVIII
6" M. I. T. G. S. Engine
Detonation Data
Fuel - "White Gas"

21 March 1950 (orifice diameter 0.920")

RPM	s	Spark Advance	Roto-meter	$\dot{M}_F$	ΔP Orifice	$\dot{M}_a$	F	Dyn. Scale	BMEP	T_1	T_{cyl}	T_{wj}	T_{air}	T_{oil}	P_1	P_e	P_a
1000	1200	15.5	5.2	.00279	7.85	.0385	.0727	15.2	59.1	180	485	179	80	151	23.05	32.05	30.1
"	"	16	5.2	.00279	7.85	.0385	.0727	15.2	59.1	"	490	180	"	150	23.05	32.05	"
"	"	22	4.9	.00267	7.0	.0364	.0732	14.7	57.2	"	490	"	"	"	22.1	31.85	"
"	1200	27	4.6	.00254	6.35	.0346	.0734	13.75	53.5	"	500	"	"	"	21.4	32.15	"
"	"	29.5	4.3	.00241	5.80	.0333	.0727	13.1	51.0	"	490	"	"	"	20.7	32.0	"
"	"	37.5	4.1	.00233	5.35	.0320	.0730	12.1	47.1	"	500	"	"	"	20.1	32.1	"
#1000	1200	31	3.95	.00225	5.0	.0309	.0729	11.65	45.3	180	475	180	80	152	19.4	32.1	30.1
# "	"	43	"	"	"	"	"	11.05	43.0	"	500	180	"	150	19.5	"	"
# "	"	47	"	"	"	"	"	10.75	41.8	"	505	182	"	"	"	"	"
# "	"	50.5	"	"	"	"	"	10.3	40.0	"	510	182	"	"	"	"	"
1000	1200	14	5.65	.00298	8.8	.0407	.0731	16.2	63.0	180	490	180	80	152	24.0	32.0	30.0
"	"	9	6.05	.00313	10.0	.0432	.0729	16.7	65.0	"	490	"	"	152	25.2	32.0	"
"	"	8	6.40	.00329	10.7	.0446	.0735	17.0	66.1	"	490	"	"	150	27.75	32.1	"
"	"	5.5	6.55	.00337	11.35	.0459	.0735	17.2	67.0	"	495	"	"	"	26.3	31.8	"
"	"	3	6.85	.00349	12.2	.0476	.0732	17.0	66.1	"	495	"	"	"	27.15	32.0	"
"	"	1	7.20	.00363	13.2	.0500	.0732	17.3	67.3	"	495	"	"	"	28.0	32.2	"
"	"	0.5	7.40	.00373	13.95	.0510	.0732	18.3	71.2	"	500	"	"	151	28.6	32.0	"

7 April 1950 (orifice diameter 0.920")

RPM	s	Spark Advance	Roto-meter	$\dot{M}_F$	ΔP Orifice	$\dot{M}_a$	F	Dyn. Scale	BMEP	T_1	T_{cyl}	T_{wj}	T_{air}	T_{oil}	P_1	P_e	P_a
1000	1200	24	8.7	.00435	10.1	.0435	.1000	18.4	71.5	180	465	180	81	150	25.3	32.0	30.1
"	"	32	8.1	.00405	8.9	.0408	.1000	17.0	66.0	"	465	"	"	152	24.1	"	"
"	"	44	7.35	.00375	7.6	.0376	.0999	14.9	58.0	"	480	"	"	151	22.8	"	"
1000	1200	30	9.5	.00470	9.8	.0427	.1100	17.7	68.9	181	445	180	81	150	25.1	32.1	30.1
"	"	38	8.95	.00445	8.8	.0406	.1095	16.6	64.5	180	445	"	"	"	24.05	32.0	"
"	"	22	10.55	.00518	11.9	.0471	.1100	20.5	79.7	180	440	"	"	"	26.9	32.0	"
1000	1200	17	7.8	.00395	10.3	.0439	.0900	18.9	73.5	181	470	181	83	150	25.55	32.0	30.1
"	"	29.5	6.6	.00342	7.7	.0379	.0901	16.05	62.5	180	470	180	"	150	22.85	"	"
"	"	38	6.2	.00323	6.9	.0359	.0900	15.2	59.1	179	80	180	"	152	22.0	"	"
1000	1200	15	6.3	.00330	9.1	.04125	.0800	17.4	67.6	180	485	180	83	152	24.4	32.0	30.1
"	"	20	5.75	.00305	7.75	.0380	.0802	16.3	63.4	180	480	"	"	152	22.9	"	"
"	"	37	4.85	.00265	5.8	.03315	.0798	13.6	52.9	179	490	"	"	150	20.6	"	"
1000	1200	17.5	5.05	.00274	7.5	.0374	.0732	15.5	60.3	181	485	180	83	150	22.65	32.0	30.1
"	"	27	4.4	.00245	6.1	.0340	.0722	14.0	54.4	180	490	"	"	"	20.95	32.2	"
"	"	35	4.2	.00235	5.55	.03245	.0724	13.15	51.2	180	495	"	"	"	20.25	32.0	"
1000	1200	21	4.25	.00238	7.2	.0367	.0650	14.0	54.4	180	480	180	83	150	22.2	32.0	30.1
"	"	28.5	3.85	.00219	6.1	.0340	.0644	12.5	48.6	180	485	180	83	150	20.9	32.1	"
"	"	37.5	3.40	.00201	5.1	.0312	.0643	11.1	43.1	"	495	"	"	"	19.7	32.1	"

INCIPIENT DETONATION OCCURRED FOR ALL POINTS EXCEPT THOSE MARKED # FOR WHICH NO DETONATION OCCURRED

TABLE XIX

Data from P-V Diagrams

transferred from

Indicator Diagrams

N(rpm)	s(ft/min)	SA(°BTC)	Area(in)²	Imep(psi)	Bmep(psi)	Fmep(psi) #
2 1/2" Engine						
960	480	35	5.362	107.24	74.7	32.5
1800	900	47	5.510	110.20	77.9	32.3
2400	* 1200	45	5.957	119.14	77.2	30.9
3000	1500	50	5.930	118.60	73.6	45.0
4" Engine						
900	720	32	5.385	107.7	91.1	16.6
1500	* 1200	39	5.720	114.4	91.8	22.6
1800	1440	41	5.780	115.6	92.5	23.1
6" Engine						
400	480	20	5.40	108.0	90.5	17.5
800	960	30	5.64	112.8	100.0	12.8
1000	* 1200	30	5.89	117.8	100.0	17.8
1200	1440	34	6.22	124.4	95.0	29.4
1500	1800	40	5.73	114.6	89.2	25.4

All indicator runs for 100 Octane fuel, 100 lb. spring, p_i = 28" Hg, p_e = 32" Hg, T_i = 150°F, T_{wj} = 180, T_{oil} = 150, at BPSA.

* Plotted in Fig. 9

Actual differences rather than differences obtained from smooth curves.

APPENDIX A

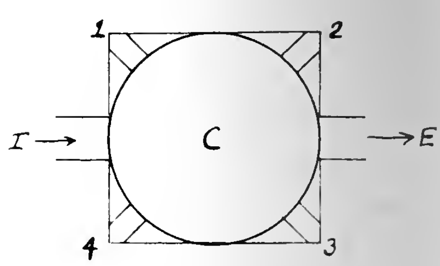
Description of Engines

General	6"	4"	2 1/2"
Bore (in)	6.0	4.0	2.5
Stroke (in)	7.20	4.8	3.0
Piston Area (in^2)	28.27	12.57	4.91
V_d (in^3)	203.5	60.35	14.71
Compression ratio	5.74	5.74	5.74
Inlet valve (in) clearance (cold)	.012	.008	.005
Exh. valve clearance (cold)	.015	.012	.006
Piston speed/rpm	1.2	.8	.5
Spark plug, champion	J7-18mm	J8	Y-4A
Valve overlap (degrees)	30	30	30
Dynamometer			
Scale piston diam (in)	2.795	1.614	.932
Dyn. torque arm (in)	21.008	15.756	12.605
Overall dyn. constant K	1000	4000	15000
Scale force for 1"Hg (lb)	3	1.0	.333
BMEP	3.89 h	3.28 h	3.59 h

$$BMEP = \frac{792,000\ h}{K\ \ V}$$

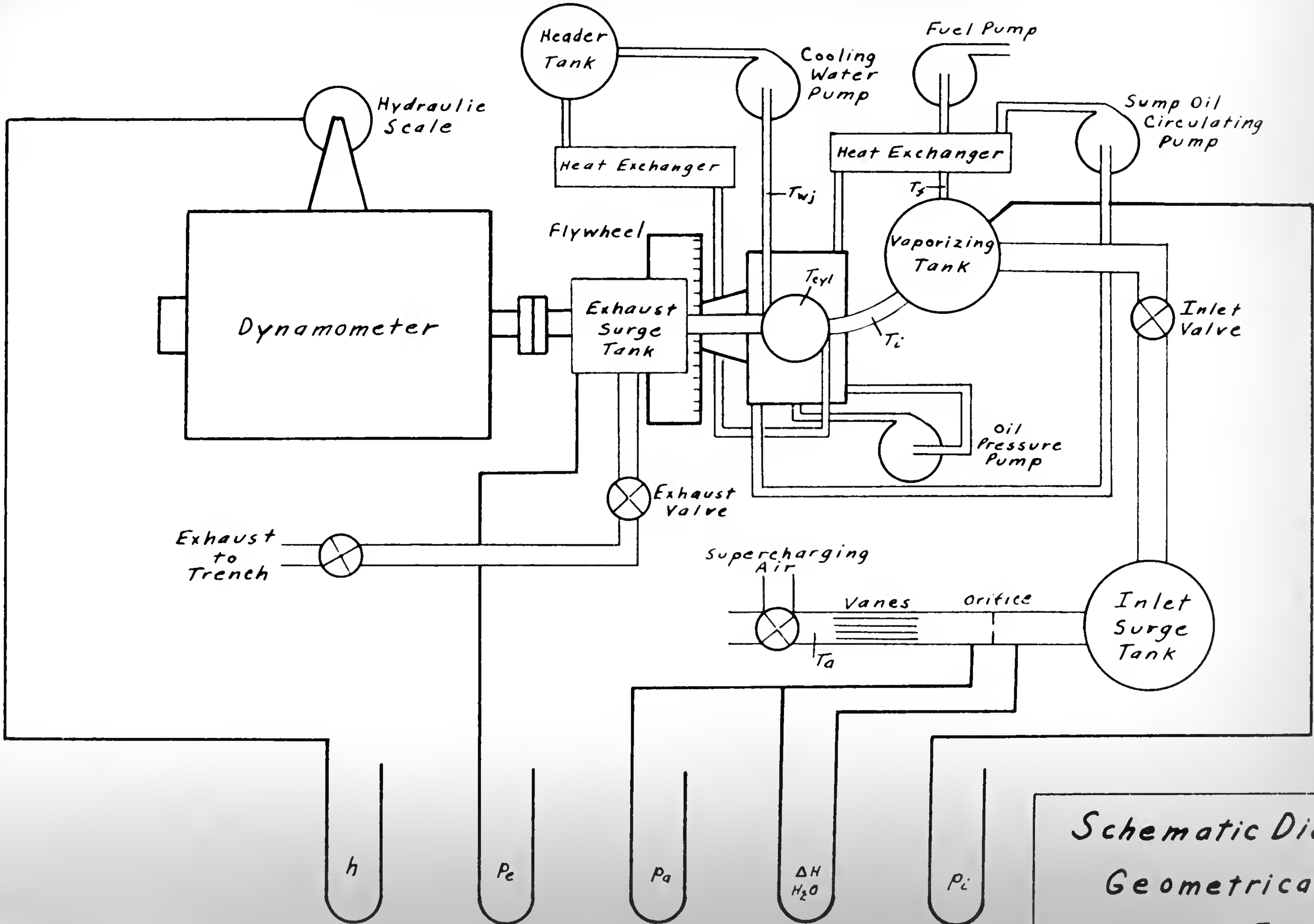
$$BHP = \frac{N\ h}{K}$$

Plug Locations

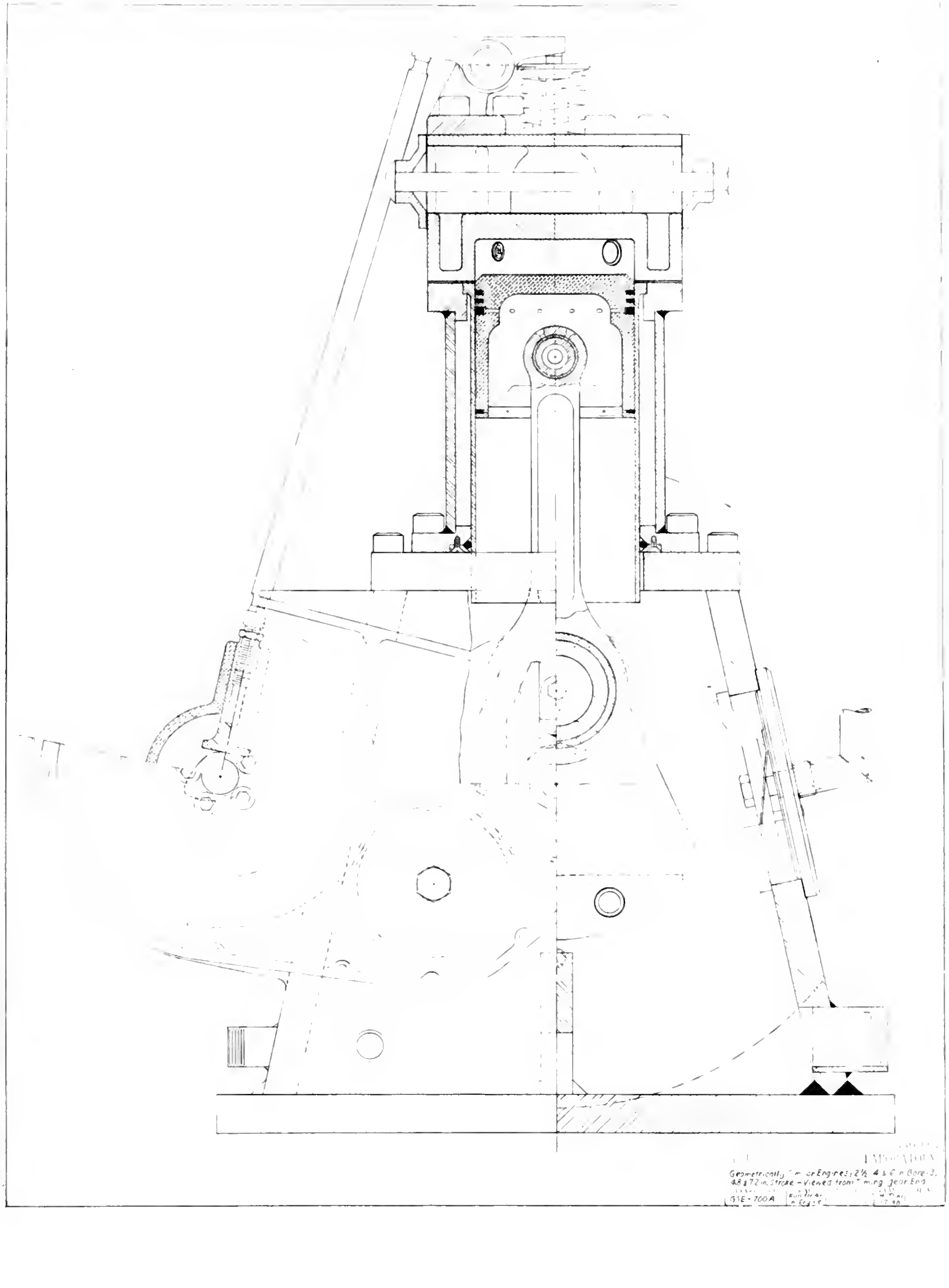


I - Inlet   C - Cylinder   *E* - Exhaust

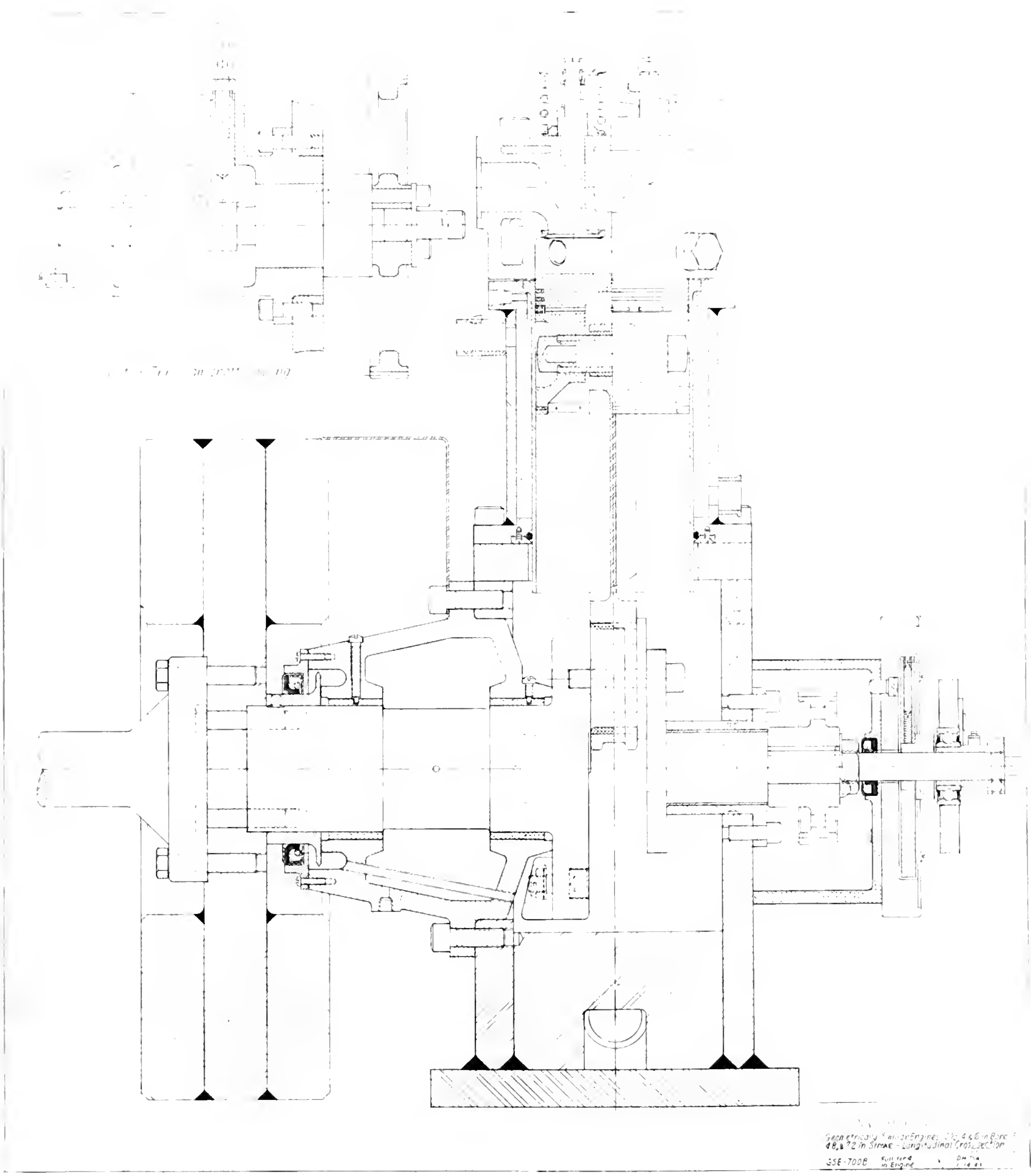
1. M.I.T. and Li Indicator
2. Non firing spark plug
3. Blind plug
4. Firing plug



Schematic Diagram
Geometrically
Similar Engines
Fig. A-1



Geometrically ... Engines; 2½ 4 & 6 in Bore-3, 4.8 & 7.2 in. Stroke - Viewed from ... ing Gear End
GSE-700A



GSE-700B
in Engine

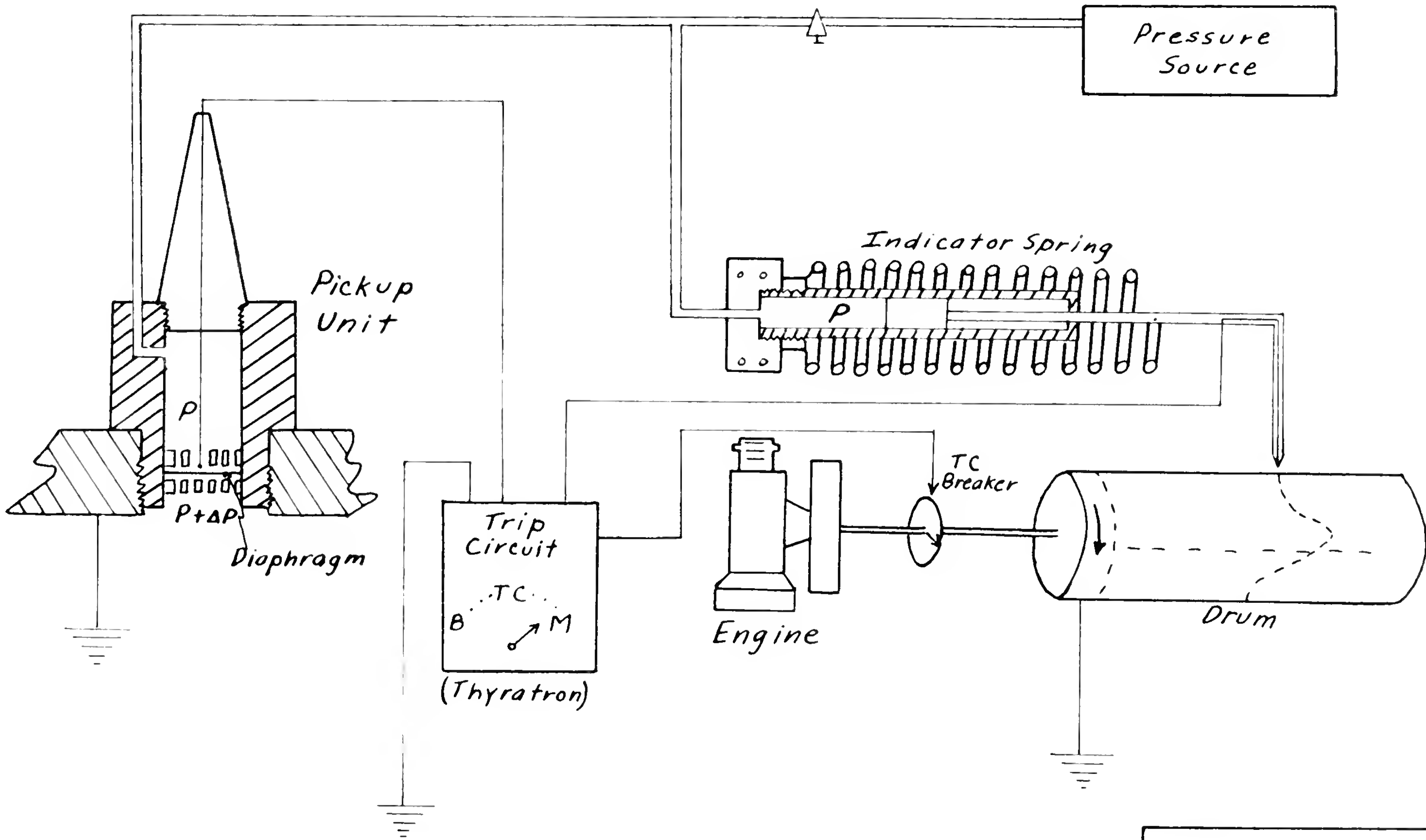
APPENDIX B

Special Equipment

The MIT indicator is a balanced pressure diaphragm type of indicator with recorder which is driven by a crankshaft coupling. The pick-up unit consists of a special plug with pressure tap and spark lead. Movement of a stylus, horizontally and parallel to a rotating drum, is controlled by a calibrated spring which extends in response to hydraulic oil pressure. A spark from the stylus to the rotating drum occurs when the hydraulic pressure in the system, is equal to the gas pressure in the engine cylinder. Establishment of an accurate top center line on the card is of utmost importance. This instrument, shown in the following Figure B1, is standard equipment in the Sloan Laboratory at MIT. It is used to give a pressure-crank angle diagram which can be converted to a p-v diagram by calculation and point to point measurement, or by a transfer machine designed to do the same process. The latter was used in this investigation and is completely described in Reference 9. Further description of the balanced pressure diaphragm unit used with the MIT indicator can be found in References 10 and 11.

The Li pressure indicator consists of a special cylinder plug and oscilloscope. The plug contains a catenary diaphragm-strain generating tube and pressure receiver. A lead from the plug is connected to the oscilloscope through an external coupling system. A condenser

connedted across the input and ground terminals of the oscilloscope differentiates the signal output of the cylinder plug unit and produces a dp/dt trace on the oscilloscope. Removal of this condenser gives a normal pressure-crank angle trace. The plug is air cooled by a compressed air line which vents the plug through radially drilled holes. Reference 12 gives a more complete description of the Li Indicator.



Schematic Diagram
MIT
Indicator
Fig. B-1

APPENDIX C

Theory for Geometrically Similar Engines

Air Capacity

$$\dot{M}_a = n\,\rho_i\,V_d\,e$$

Since in G. S. Engines, $V_d \sim \ell^3$

and since $\ell\,n \sim s$

$$\dot{M}_a \sim s\,\ell^2 \rho_i\,e$$

Therefore, at same inlet conditions and piston speed

$$\dot{M}_a \sim \ell^2 e$$

Volumetric efficiency

Examine now all variables on which e is dependent.

$$e = e(N, \ell, \rho_i, \mu_i, c_i, R_1, R_2 \cdots R_n)$$

Where R_1, R_2......R_n are design ratios and are the same for G. S. Engines.

$$\therefore\ e = e(N, \ell, \rho_i, \mu_i, c_i)$$

The above relation contains five variables. According to the Buckingham Pi Theorem there should be 2 independent dimensionless ratios, with 3 independent dimensions.

Buckingham Pi Theorem

M = unit of mass; L = unit of length; T = unit of time

Variable	Units	Exponent
N	T^{-1}	a
ℓ	L	b
ρ_i	ML^{-3}	c
μ_i	$ML^{-1}T^{-1}$	d
c_i	LT^{-1}	e

Basic Equation Becomes

$$(T^{-1})^a (L)^b (ML^{-3})^c (ML^{-1}T^{-1})^d (LT^{-1})^e = T^o L^o M^o$$

(1) $T^o = T^{-a-d-e}$

$$-a-d-e = o$$

(2) $L^o = L^{b-3c-d+e}$

$$b-3c-d+e = o$$

(3) $M^o = M^{c+d}$

$$c+d = o$$

Solving simultaneously (1), (2) and (3),

$$\frac{b}{e} - 2\frac{a}{e} = 1$$

As is seen from this relation there are only two independent values of the above variables $\frac{b}{e}$ and $\frac{a}{e}$. All other values of these variables may be expressed in terms of the two independent values chosen.

Let $\frac{a}{e} = 1$ then $\frac{b}{e} = 3$ (1)

$\frac{a}{e} = 2$ then $\frac{b}{e} = 5$ (2)

Using Ratio (1): $\frac{a}{e} = 1, \ \frac{b}{e} = 3.$

$$\frac{d}{e} = -2$$

$$\frac{c}{e} = 2$$

Using Ratio (2) $\frac{a}{e} = 2, \ \frac{b}{e} = 5$

$$\frac{d}{e} = -3$$

$$\frac{c}{e} = +3$$

Assigning exponent $e = 1$, since interest is only in the lowest ratio of exponents, the two independent dimensionless ratios are obtained:

$$\pi_1 = N^1 \ell^3 \rho_i^2 \mu_i^{-2} c_i^1 = \frac{N \ell^3 \rho_i^2 c_i}{\mu_i^2}$$

$$\pi_2 = N^2 \ell^5 \rho_i^3 \mu_i^{-3} c_i^1 = \frac{N^2 \ell^5 \rho_i^3 c_i}{\mu_i^3}$$

Further manipulation of π_1 and π_2 to obtain simpler π's, yields:

$$\pi_1' = \frac{N \ell^2 \rho_i}{\mu_i} \qquad \pi_2' = \frac{N \ell}{c_i}$$

Since $\ell N \sim s$, and if s is accepted as the characteristic velocity of the inlet process, it is seen that:

$$\pi_1' \sim \frac{\rho_i s \ell}{\mu_i} \sim \text{Reynolds Number}$$

$$\pi_2' \sim \frac{s}{c_i} \sim \text{Mach Number}$$

Therefore, from the preceding derivation

$$e = e(\text{Reynolds Number, Mach Number})$$

Power Output

$$IMEP = \frac{JFE_c \dot{M}_a \eta_i}{144\, n\, V_d}$$

$$\dot{M}_a = n \rho_i V_d\, e$$

$$IMEP = JFE_c \eta_i \frac{n \rho_i V_d\, e}{144\, n\, V_d}$$

$$IMEP = e \frac{JFE_c}{144} \rho_i \eta_i$$

If in G. S. Engines, η_i is the same at the same F (and at best power spark advance), then for the same inlet conditions

$$IMEP \sim e \sim e(R_e, M)$$

$$IHP = \frac{IMEP\ V_d\ N}{2 \times 12 \times 33000} \sim IMEP\, \ell^3 N$$

$$IHP \sim IMEP\, \ell^2 s \sim e\, \ell^2 s$$

In comparing HP output at the same s, $\frac{IHP}{\ell^3} \sim \frac{e}{\ell}$

Since the weight of an engine is proportional to ℓ^3

$$\frac{IHP}{wt} \sim \frac{e}{\ell}$$

Inertia Stresses

Since the cylinder gas pressure and the thermal stresses do not vary radically over the operating range of an engine, the inertia stresses of the moving parts are of prime importance

in determining the maximum speed of an engine or in operating various engines at the same stress condition.

$$\sigma_{in} \sim \frac{MASS\ \omega^2}{l} \sim \frac{l^3 \omega^2}{l}$$

since $l\omega \sim s$

$$\sigma_{in} \sim s^2$$

FRICTION

a) Viscous Friction

Petroff's equation for full film lubrication:

$$FHP = 3.80 \times 10^{-13}\ \frac{D}{C}\ \frac{L}{D} \mu N^2 D^3$$

In G. S. Engines $\frac{D}{C}$, $\frac{L}{L}$ will be the same; $FHP \sim \mu N^2 l^3$

Since $lN \sim s$

$$FHP \sim \mu s^2 l$$

Also, $FHP \sim FMEP\ \ V_d\ \ N \sim FMEP\ l^2 s$

Therefore $FMEP \sim \frac{\mu}{l} s$

and at the same s:

$$FMEP \sim \frac{\mu}{l}$$

B. Coulomb Friction

$$FMEP \sim \frac{f\cdot\left(\begin{smallmatrix}\text{load}\\ \text{pressure}\end{smallmatrix}\right) l^2 l}{V_d} \sim f\cdot\left(\begin{smallmatrix}\text{load}\\ \text{pressure}\end{smallmatrix}\right)$$

Coulomb FMEP is therefore proportional to the product of some coefficient of friction and a load pressure. In

G. S. Engines, f should be the same since it is primarily dependent on type of material, surface, etc.; and at the same s, the gas pressure or any load pressure should be the same. Therefore, the Coulomb FMEP should be the same for G. S. Engines.

C. Pumping Loss

$$PMEP = \frac{\text{Exhaust Stroke Work - Inlet Stroke Work}}{V_d}$$

$$PMEP = (p_e + \overline{\Delta p}_e)\frac{V_d}{V_d} - (p_i - \overline{\Delta p}_i)\frac{V_d}{V_d}$$

where $\overline{\Delta p}_e$ and $\overline{\Delta p}_i$ represent a mean pressure difference above and below exhaust and inlet pressures respectively, and are due to the dynamic effects of gas flow out of and into the cylinder.

Therefore:

$$\Delta p_i \sim \rho_i U_i^2 \qquad \text{and}$$

$$\overline{\Delta p}_i = \frac{1}{V_d}\int \Delta p_i \, dv$$

$$\overline{\Delta p}_i = \frac{1}{V_d}\int \rho_i U_i^2 \, dv$$

A similar argument holds for $\overline{\Delta p}_e$.

If in G. S. Engines the inlet and exhaust values are similar in that the gas velocities are the same at the same piston speeds, the PMEP due to $\overline{\Delta p}_e$ and $\overline{\Delta p}_i$ (dynamic effects) should be the same. If the same inlet and exhaust pressures

prevail, then the total PMEP of G. S. Engines should be the same at the same s.

APPENDIX D

Fuels and Lubricants

In the friction investigation 100/130 aviation fuel was used, while in the detonation runs white marine gasoline (unleaded) was used. This lower octane fuel (71.9 motor method, 78.4 research method) produced detonation in all engines in the range of speeds and inlet pressures investigated.

Similitude was carried out in the respective lubricating oil viscosities by using the specially prepared oils listed in the following table.

	Viscosity s.s.u.			
Engine	100°F	130°F	210°F	Gravity
2 1/2"	349	160.3	52.8	.88
4"	828	339.0	76.2	.89
6"	1665	627.0	114.1	.90

The oils were prepared using SAE 20 and SAE 60 as follows:

2 1/2"	URSA P 20 (Texaco symbol)
4"	54% URSA P 60, 46% P 20
6"	95% URSA P 60, 5% P 20

Each oil was a straight mineral oil, wholly paraffin and distilled, with no additives.

The above samples give the same value of μ/ρ at 250°F. This temperature was arbitrarily selected as an attempt to satisfy lubrication requirements in all parts of the engine where viscous friction may occur. A mean oil crankcase inlet temperature of 150°F was chosen and maintained constant throughout the investigation.

[illegible]

[illegible]

Air flow in all engines was measured by means of the ASME square-edged orifice with flange taps, as described fully in Reference 13. The flange taps are static pressure taps located 1 inch from the upstream and downstream faces of the orifice.

The equation for the flow rate of a gas through a square-edged orifice may be written as follows:

$$\frac{w}{g_0} = A \frac{C}{(1-\beta^4)^{1/2}} Y_1 \left[2 \frac{\rho_1}{g_0} (p_1 - p_2) \right]^{1/2} \qquad (1)$$

where

w = flow rate, lb/sec

A = orifice area, ft^2

C = discharge coefficient, dimensionless

β = D_2/D_1 = ratio of orifice diameter to pipe inside diameter, dimensionless

ρ_1 = density of the gas before the orifice, lb/ft^3

p_1 = static pressure ahead of the orifice, lb/ft^2 abs.

p_2 = static pressure after the orifice lb/ft^2 abs.

g_0 = acceleration of a unit mass when acted on by a unit force, ft/sec^2

Units of force, mass and acceleration are as follows:

$$\text{Force} = \text{mass} \times \text{acceleration}$$

$$1 \text{ pound force} = \frac{1 \text{ lb.mass}}{g_o} \times 1 \text{ ft/sec}^2$$

Orifice Area

Expressing A, orifice area, in terms of the diameter of the orifice,

$A = \frac{\pi D_2^2}{4} = 0.7854 D_2^2$, ft^2 where D_2 is expressed in ft.

or $A = .005454 D_2^2$, ft^2 where D_2 is expressed in inches.

Discharge Coefficient

C is the discharge coefficient and is defined as:

$$C = \frac{\text{actual mass rate of flow}}{\text{theoretical mass rate of flow}}$$

Values of C for standard orifices have been determined by the ASME through experiment.

Velocity of Approach Factor

$(1-\beta^4)^{1/2}$ is the velocity of approach factor and must be used if p_1 is measured by a static tube. If p_1 is measured by an impact tube (i.e. if total pressure is measured), the approach factor is unity. Also, for small values of β, this factor is negligible.

Expansion Factor

Y_1 is the expansion factor and accounts for the uncontrolled expansion (both longitudinally and laterally) of a compressible fluid after the orifice. Y_1 has been determined experimentally, and empirical equations have been fitted to the data. The equation for air is:

$$Y_1 = 1 - \frac{(p_1 - p_2)}{p_1}(0.41 + .35\beta^4)$$

It is seen that for constant K and β, Y_1 varies linearly WITH $\frac{p_1 - p_2}{p_1}$. A plot of the above relation for various β's is available in Ref. 13.

Density

The density of dry air at 520°F abs. and 14.70 lb/in^2 pressure is .07637 lb/ft^3. Therefore the density at any temperature and pressure can be expressed as follows:

$$\rho = .07637 \frac{p}{14.70} \frac{520}{T} = 2.702 \frac{p}{T}, \frac{lb}{ft^3}$$

where p = pressure, lb/in^2 abs.

T = temp., deg. F abs

If the pressure p is measured in inches Mercury,

$$\rho = 1.321 \frac{p}{T}, \frac{lb}{ft^3}$$

p = pressure, in. Hg abs.

T = temp., deg. F abs.

To account for deviations of the above relation from the perfect gas law, a correction factor could be applied.

The magnitude of this factor, within the range of pressures and temperatures encountered in this investigation, does not warrant inclusion.

Orifice Pressure Drop

The pressure drop $(p_1 - p_2)$ across the orifice is measured in inches of water.

$(p_1 - p_2) = 5.188\ h,\ lb/ft^2$ where

h = pressure drop across orifice, inches water.

Flow Coefficient

The discharge coefficient and velocity of approach factor are combined into a single dimensionless expression, K. This flow coefficient K is defined as:

$$K = \frac{C}{(1-\beta^4)^{1/2}}$$

Experimental values of K as a function of Reynolds Number, β and D_1 have been compiled by the ASME, and are available in Ref. 13 .

Working Equation

Equation (1) can now be written in terms of the symbols defined and the units by which measurements were made:

$$\frac{w}{32.17} = .005454 D_2^2\ KY_1 \left[2\ \frac{1.321}{32.17}\ \frac{p_1}{T_1}\ (5.188\ h\right]^{1/2}$$

or $$w = .1145 D_2^2\ KY_1 \left[\frac{p_1}{T_1}\ h\right]^{1/2} \qquad (2)$$

w = flow rate of air, lb/sec

D_2 = orifice diameter, inches

p_1 = static pressure ahead of the orifice, inches Hg abs.

T_1 = temperature ahead of the orifice, deg. F abs.

h = pressure drop across the orifice, inches of water

K = flow coefficient = function of Re, β, D_1

Y_1 = expansion factor = function of $\frac{\Delta p}{p_1}$, β

Air flow, in these investigations was calculated from the above formula. It was found that by using values of K and Y_1 corresponding to mean values of Re = 50,000 and $\frac{\Delta p}{p_1}$ = .035, respectively, a maximum error of ±3% would be encountered at extremely low and high flow rates. Since Re can be expressed in terms of w, μ, and D_2; and $\frac{\Delta p}{p_1}$ in terms of w, D_2, p_1 and T_1; a correction to the original value of w, based on a mean Re and $\frac{\Delta p}{p_1}$ as a function of w itself, was possible. Such a correction curve was computed for each orifice for the p_1 and T_1 encountered. The curve for the 2 1/2" engine, D_2 = .413, β = .20, p_1 = 29.92 in. Hg., T_1 = 30°F is included as Fig. E-1.

<u>Computing Reynolds Number for Correction Curve</u>

Reynolds Number is defined as

$$R_e = \frac{\rho U D_2}{\mu}$$

ρ = fluid density ahead of the orifice, lb/ft³

U = velocity through the orifice, ft/sec

D_2 = diameter of the orifice, inches

μ = viscosity of air before the orifice, $\frac{\text{lb.mass}}{\text{ft.sec}}$

From continuity:

$$u = \frac{w}{\rho A} = \frac{w}{\rho \frac{\pi}{4}\left(\frac{D_2}{12}\right)^2} = \frac{576\, w}{\rho \pi D_2^2}$$

Substituting this expression for u in the expression for Re,

$$Re = \frac{576\, w}{\rho \pi D_2^2} \; \frac{\rho D_2}{12 \mu} = \frac{15.28\, w}{D_2 \mu}$$

Computing $\frac{\Delta p}{p_1}$ for correction curve

Eq(2)

$$w = .1145\, D_2^2\, K_m Y_{1m} \left[\frac{p_1 h}{T_1}\right]^{1/2}$$

$$h = \left[\frac{w}{.1145 D_2^2 K_m Y_{1m}}\right]^2 \frac{T_1}{p_1}\text{, in. water}$$

$$\frac{h}{p_{1_{in.}H_2O}} = \left[\frac{w}{.1145 D_2^2 K_m Y_{1m}}\right]^2 \frac{T_1}{p_1} \times \frac{1}{p_{1_{in}H_2O}}$$

2 1/2" G.S.E.

AIRFLOW CORRECTION CHART
ORIFICE DIAM. 0.413"
$T_1 = 80°F$ $p_1 = 29.92$ "Hg $\beta = .20$

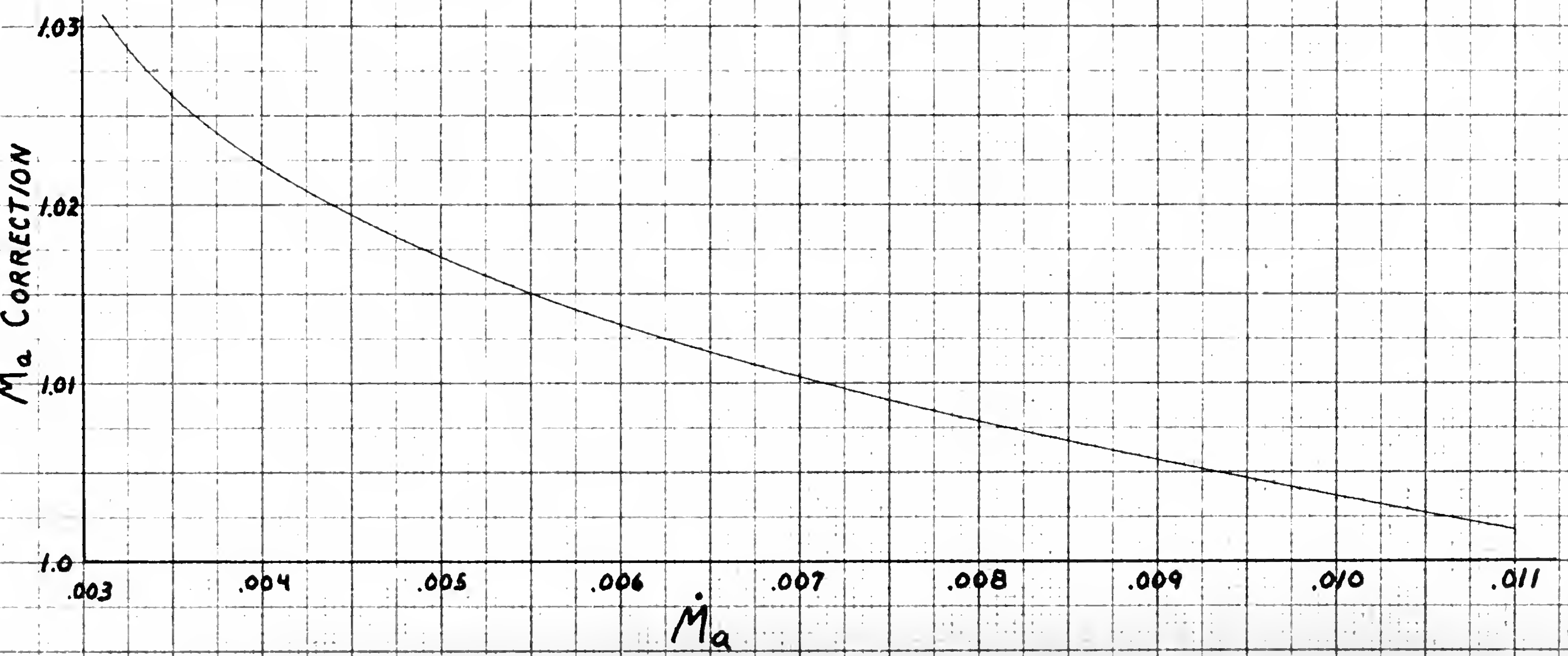


FIG. E-1

APPENDIX F

FUEL FLOW MEASUREMENT

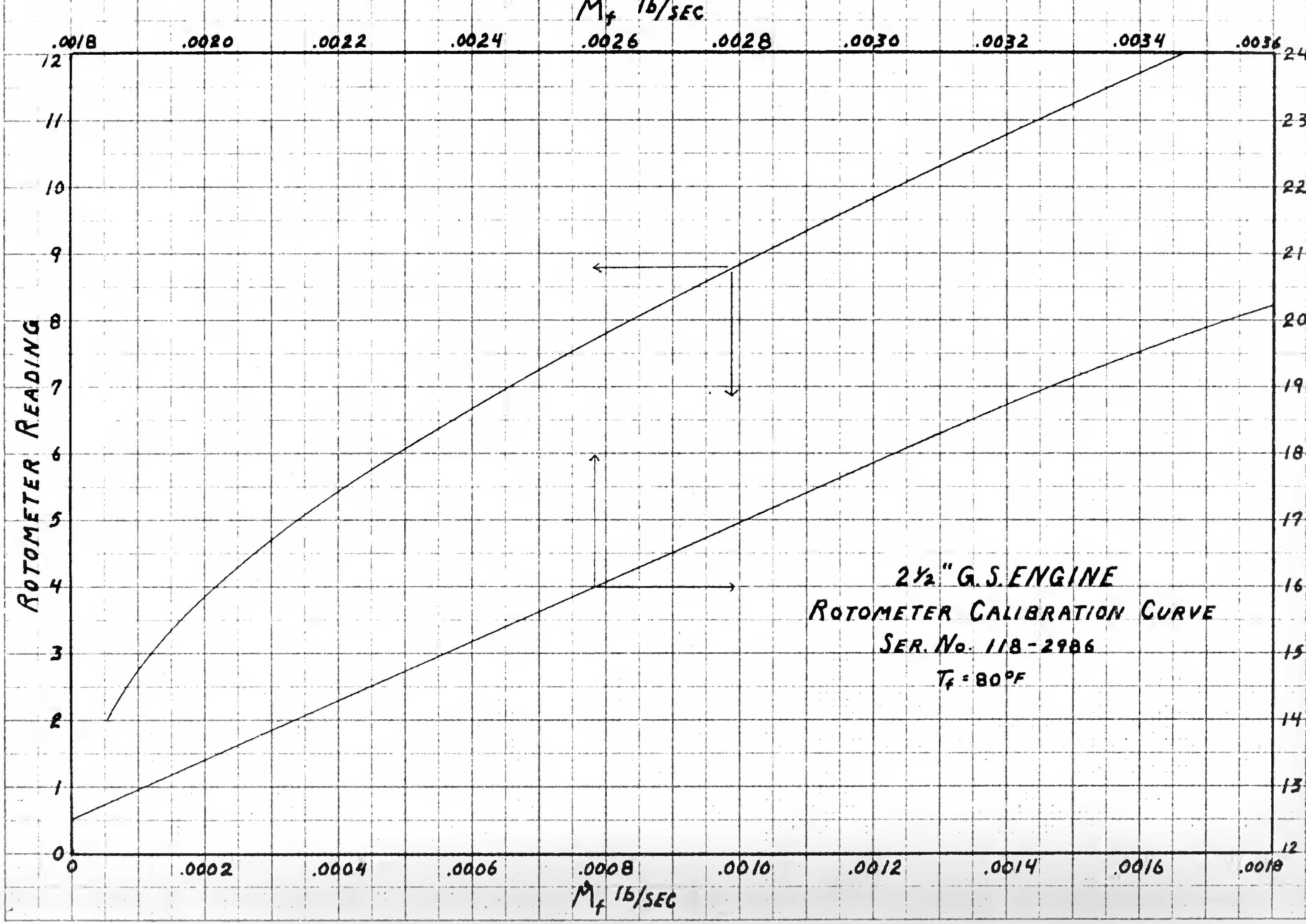
The flow rate metering instruments, or rotometers, used throughout this investigation for the measurement of fuel flow, operated according to the laws governing the flow of fluids through apertures. Specifically, the fluid was discharged under controlled head conditions through an annular aperture of controlled variable size. The actual rotometers used were as follows:

2 1/2" engine - Fischer and Porter, No. 118-2986

4" engine Large-Fischer and Porter, No. H8-2992
Small " " " " A-65114, Fig. No. 12

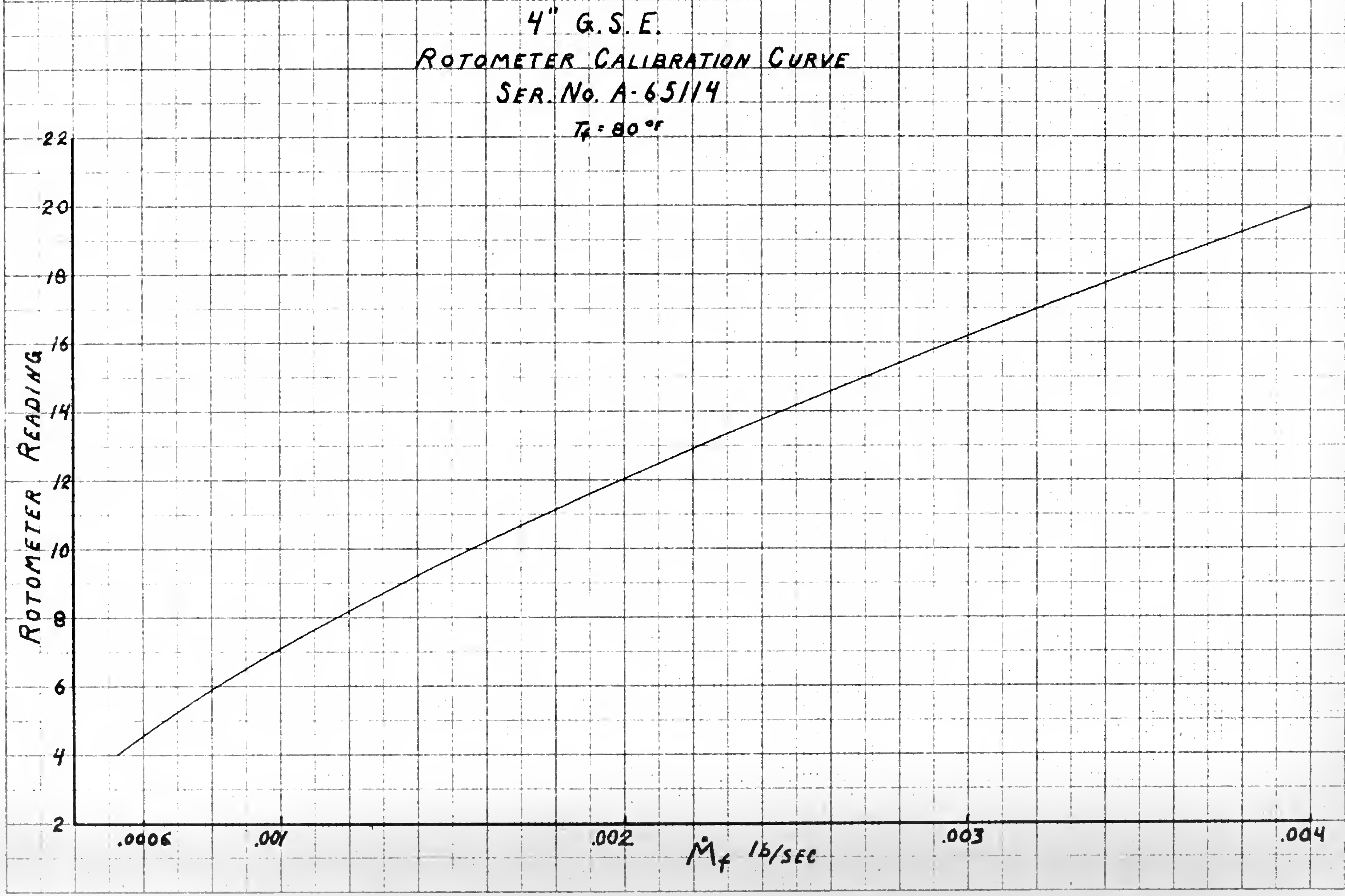
6" engine - Fischer and Porter, No. 118-2996

All rotometers were calibrated for both 100 octane gasoline and 71.9 octane (motoring method) unleaded gasoline. There was no appreciable difference in the calibration results for the two fuels. The calibration consisted of determining, by an electric timer, the time required for a given weight of fuel to flow into a container at a constant rate of flow. The rate of flow was controlled through the rotometer, and the rotometer reading was noted for each flow rate. Thus, the full range of flow for each rotometer was checked in this manner. The rate of flow in pounds per second was plotted versus rotometer reading to give the calibration curves, Figs. F-1, F-2, F-3. For further discussion of the theory of the subject rotometer, see Ref. 14.



$\dot{M}_f$ lb/sec
.0018
.0020
.0022
.0024
.0026
.0028
.0030
.0032
.0034
.0036
Rotometer Reading
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
19
20
21
22
23
24
2½" G.S. ENGINE
Rotometer Calibration Curve
Ser. No. 118-2986
T_f = 80°F
.0002
.0004
.0006
.0008
.0010
.0012
.0014
.0016
.0018
$\dot{M}_f$ lb/sec

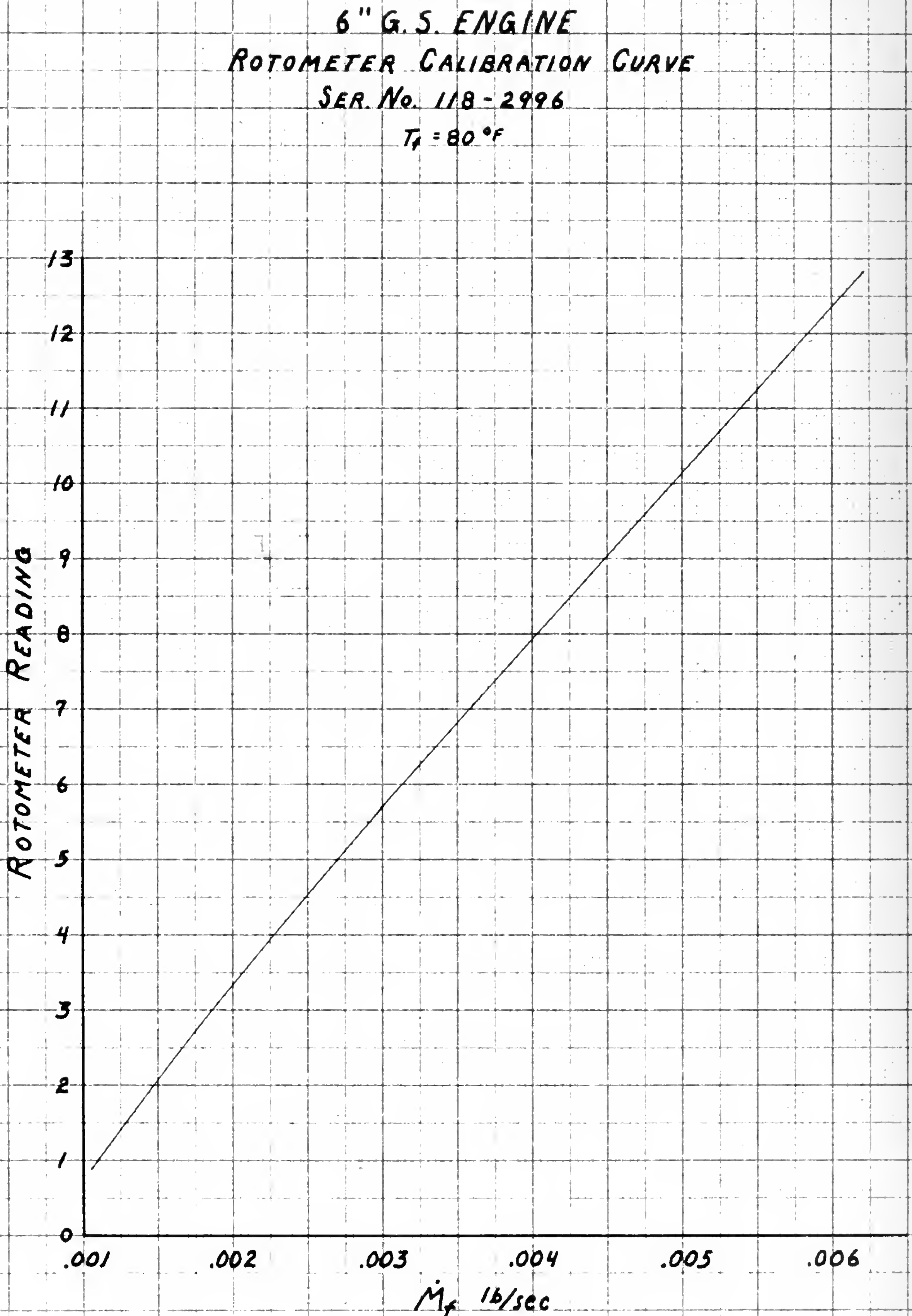
FIG. F-1



4" G.S.E.
ROTOMETER CALIBRATION CURVE
SER. No. A-65114
$T_f = 80°F$
ROTOMETER READING
22
20
18
16
14
12
10
8
6
4
2
.0006
.001
.002
.003
.004
$\dot{M}_f$ lb/sec

FIG. F-2

Fig. F-3


6" G.S. ENGINE
ROTOMETER CALIBRATION CURVE
SER. No. 118-2996
T_f = 80 °F
ROTOMETER READING
0
1
2
3
4
5
6
7
8
9
10
11
12
13
.001
.002
.003
.004
.005
.006
$\dot{M}_f$ lb/sec

Thesis 12990
G12 Gaboury
A study of friction and detonation in geometrically similar engines.